ENGINEERING DESIGN HANDBOOK

AUTOMOTIVE SERIES

AUTOMOTIVE BODIES AND HULLS

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LIST OF SYMBOLS

A = general symbol for area, ft^2 (also $in.^2$); effective cross-sectional area of a weld (area of maximum stress), $in.^2$; surface area in contact with fluid, ft^2; maximum amplitude of vibration, in.; area perpendicular to heat flow, ft^2

A_{CF} = cross-sectional area of compression flange of stressed-skin panel, $in.^2$

A_{TF} = cross-sectional area of tension flange of stressed-skin panel, $in.^2$

A_b = projected bearing area, $in.^2$

A_i = effective cross-sectional area of any weld segment in a pattern (area of maximum stress), $in.^2$

A_u = ordinary cross-sectional area of stiffener of stressed-skin panel, exclusive of skin, $in.^2$

A_{ue} = effective cross-sectional area of stiffener of stressed-skin panel, exclusive of skin, $in.^2$

a = general symbol for acceleration, ft/sec^2; metacentric height, ft; horizontal distance from vehicle CG to the line of action of the vertical bank reaction force S, ft

B = breadth (width) of vehicle, ft

$B_1, B_2, ...B_6$ = constants used with Eqs. 4-50 to 4-53 and given in Table 4-7

b = width of a column, in.; unsupported width of skin of a stressed-skin structure, in.; horizontal distance from vehicle CG to line of action of buoyancy force F_B, ft

b_1 = pitch (spacing) of skin stiffeners of stressed-skin panels, in.

b_e = effective width of skin and stiffener that resist buckling in a stressed-skin panel, in.

C = deflection coefficient, dimensionless; proportionality constant used in calculations involving eccentrically loaded groups of fasteners, lb/in.

$\bar{C}$ = length of moment arm of center of gravity moment about center of roll, ft

C' = proportionality constant used in calculations involving eccentrically loaded welded joints, $lb/in.^3$

C_1 = angle correction factor relating to diagonal tensile forces in stressed-skin panels, dimensionless

C_2 = stress concentration factor relating to increased stresses at corners of stressed-skin panels, dimensionless

C_3 = stress concentration factor relating to increased stresses at localized areas of flanges of stressed-skin panels, dimensionless

C_D = total drag coefficient, dimensionless

CB = center of buoyancy, dimensionless

CB_1, CB_2 = centers of buoyancy of submerged volumes 1 and 2, respectively

CG = center of gravity, dimensionless

D = diameter of rivet head, in.; flexural rigidity in plates, lb-in.

d = body diameter of rivet, bolt, or screw, in.; a parameter of stressed-skin structures, dimensionless; draft at bow, ft

LIST OF SYMBOLS (Cont'd)

d_i = inside diameter, in.

d_m = mean diameter, in.

E = modulus of elasticity, lb/in.^{2}; dissipator thickness efficiency

EHP = effective horsepower

E_c = modulus of elasticity of core material of sandwich panel, lb/in.^{2}

E_f = modulus of elasticity of facing skin material of sandwich panel, lb/in.^{2}

e = eccentricity; in a stressed-skin panel, it is the distance from the centroid of the stiffener cross section A_u to the mid-thickness of the skin, in.; distance of vehicle CG from bow, ft

F = general symbol for force, lb

F_B = buoyancy force, lb

F_{B_1}, F_{B_2} = buoyancy force of volumes 1 and 2, respectively, lb

F_D = total drag force, lb

F_S = shearing force, lb

F_b = bearing load, lb

F_d = direct load component of the total load on a fastener (rivet, bolt) of a fastener group loaded eccentrically, lb

F_i = moment load component on any one fastener of an eccentrically loaded group of fasteners, lb

F_ℓ = Froude number = $V/\sqrt{g\ell}$, dimensionless

F_x = force component in x direction, lb

f_b = factor of safety for bearing stress, dimensionless

f_i = combined effects coefficient of convection and radiation at the higher temperature wall surface, Btu/hr-ft^2-°F

f_o = combined effects coefficient of convection and radiation at the lower temperature wall surface, Btu/hr-ft^2-°F

f_s = factor of safety for shear stress, dimensionless

G = modulus of rigidity, lb/in.^{2} (also known as shear modulus); deceleration rate, g's; specific gravity of wood, lb/in.^{3}

G_c = modulus of rigidity (shear modulus) of core material of sandwich panel, lb/in.^{2}

g = acceleration due to gravity, ft/sec^2

H = steady state heat flow, Btu/hr; distance between centroids of stressed-skin structure, in.; maximum height of heaped cargo, in.; height of vehicle, ft

$\overline{H}$ = length of moment arm of buoyancy moment about center of roll, ft

h = film coefficient of heat transfer, Btu/hr-ft^2-°F; unsupported height of skin of a stressed-skin structure, in.; height from which a load is dropped to produce a dynamic load on cargo floor, in.; basic size of a weld; in a fillet weld, it is the length of one leg of the fillet; in a "V" weld, it is the depth of the "V", etc., in.

$h'_{(max)}$ = maximum depth of immersion of a wall into a fluid medium, in.

h_1 = inside height of cargo box sides, in.

h_2 = maximum height of heaped cargo above cargo box sides, in.

h_r = height of wall reinforcement in cargo or dump body, in.

LIST OF SYMBOLS (Cont'd)

h_u = free length of stiffener in a stressed-skin panel, in. (distance between centroid of rivet pattern or between welds)

I = mass moment of inertia, ft-lb-sec^2 or slug-ft^2; moment of area of a section about a transverse centroidal axis, in.^{4}

I_{CF} = moment of inertia of compression flange of stressed-skin panel about its neutral axis, in.^{4}

I_F = moment of inertia of flange cross section about beam neutral axis of stressed-skin panel, in.^{4}

I_g = moment of inertia of vehicle about longitudinal axis through CG, slug-ft^2

I_o = mass moment of inertia about axis O, ft-lb-sec^2 or slug-ft^2

I_T = moment of inertia of transverse cross section of cylindrical tank, in.^{4}

I_{TF} = moment of inertia of tensile flange of stressed-skin panel about its neutral axis, in.^{4}

J = general symbol for the polar moment of inertia of a section, in.^{4}; the polar moment of inertia of the total shear area of a weld pattern about the centroid of the pattern, in.^{4}

J_T = polar moment of inertia of cross section of cylindrical tank, in.^{4}

J_{oi} = polar moment of inertia of one weld segment of a weld pattern about its own centroid, in.^{4}

K = combined radiation coefficient that includes a radiation constant, a relative-position factor for the emitting and receiving surfaces, and absorption and emission characteristics of the two surfaces, Btu/hr-ft^2-°F; coefficient of torque used in calculations of bolt loading, dimensionless

K_a = windchill, kg-cal/m^2-hr

K_1 = portion of total torque effort wasted to friction at bearing surface of nut or bolt, dimensionless

K_2 = portion of total torque effort wasted to friction at thread contact surfaces, dimensionless

K_3 = portion of total torque effort that results in useful bolt tension, dimensionless

K_s, K_s' = theoretical buckling coefficients for stressed-skin structure loaded in shear (functions of sheet dimensions and type of edge restraint), dimensionless

K_{ss} = theoretical shear buckling coefficient for sheet of height h and width b with all edges simply supported in a stressed-skin structure, dimensionless

k = spring constant, lb/in.; slope (ratio of vertical to horizontal distances) of material heaped above the cargo box, dimensionless; radius of gyration of vehicle about its center of gravity, ft; wind gust factor, dimensionless; thermal conductivity of material, Btu/hr-ft^2-°F/ft (which can be reduced to Btu/hr-ft-°F)

$k_1, k_2, k_3, \ldots k_n$ = respective thermal conductivi-

LIST OF SYMBOLS (Cont'd)

ties of materials in layers 1, 2, 3, ...n, Btu/hr-ft^2-°F/ft (which can be reduced to Btu/hr-ft-°F)

L = length of span, in. (also ft); length of vehicle, ft; length of recoil stroke, ft

L_e = effective column length of stiffeners of stressed-skin panels, in.

ℓ = length of wall, in.; length of weld, in.; waterline length of amphibious vehicle, ft; length of vehicle from front of chassis to CG

ℓ_1 = length of any weld segment in a welding pattern, in.

M = mass, lb-sec^2/ft or slugs; general symbol for moment, or bending moment, in.-lb (also ft-lb); bending moment per unit width or length of beam or plate, in.-lb/in.

M' = secondary bending moment, in.-lb

M_A = bending moment induced by acceleration, in.-lb

M_{SL} = bending moment on cylindrical tank due to side loading, in.-lb

M_T = torsional moment on cylindrical tank, in.-lb

M_g = mass of propelling gases and unburned powder, lb-sec^2/ft

$M_{(max)}$ = general symbol for maximum value of a bending moment, in.-lb; maximum bending moment per unit width or length of beam or plate, in.-lb/in.

$M'_{(max)}$ = maximum effective bending moment per unit length of wall including effect of reinforcements, in.-lb/in.

M_p = mass of projectile, lb-sec^2-ft

M_r = mass of recoiling parts, lb-sec^2-ft

MC = metacenter, dimensionless

MS_C = margin of safety in compression, dimensionless

MS_T = margin of safety in tension, dimensionless

N = total number of fasteners in a rivet or bolt group

n = basic load factor for estimating dynamic road loads, dimensionless; diagonal tension factor in stressed-skin structures, dimensionless

P = load, lb; dynamic load (equivalent static load that will produce same stress and deflection as a given falling weight, lb; peak pressure during explosion, psi

P_d = dynamic wind pressure, psf (also psi)

p = concentrated or total load acting on one-inch width of sandwich panel, lb/in. of width

Q_h = steady state heat flow due to convection, Btu/hr

Q_k = steady state heat flow due to conduction, Btu/hr

Q_r = steady state heat radiated, Btu/hr-ft^2

q = distributed unit loading, psi; unit loading per inch of width of sandwich panel, lb/in. per in. of width

$q_{(max)}$ = maximum unit loading, lb/in.^{2}

R = radius, ft; distance to explosive charge, ft; radius of curvature of curved web of a stressed-skin structure, in.

R_b, R_h = empirical coefficients relating to types of edge restraints of skin of stressed-skin structures, dimensionless

LIST OF SYMBOLS (Cont'd)

r = radius of gyration of stiffener cross section A_u about an axis at, and parallel to, the skin contact surface, in.

r_B = effective radius of friction force acting on bearing surface of threaded fastener, in.

r_i = radial distance from centroid of the total shearing area of a fastener group or weld pattern to any one fastener or point in the pattern, in.

$r_{(max)}$ = radial distance from the centroid of the total shear area of a weld pattern to the most distant point in the pattern, in.

r_{oi} = distance from the centroid of any weld segment of a pattern to the centroid of the pattern, in.

S = vertical reaction force at bank on amphibious vehicle, lb; total wetted area of hull, ft^2; normal stress, lb/in.^{2}

S' = stress due to dynamic loading, lb/in.^{2}; edge distance, i.e., the distance from edge of riveted plate to center of nearest rivet, in.

S_B = resultant torsional stress S_i experienced by point B in a given weld pattern, lb/in.^{2}

S_{Bh} = horizontal component of the torsional stress S_B experienced by point B in a given weld pattern, lb/in.^{2}

S_{Bv} = vertical component of the torsional stress S_B experienced by point B in a given weld pattern, lb/in.^{2}

S_{CB1} = compression stress due to primary bending in compression

S_{CB2} = secondary bending stress in compression flange of stressed-skin panel due to vertical component of diagonal tensile stresses in web, lb/in.^{2}

S_{CC} = average compressive axial stress in compression flange of stressed-skin panel due to diagonal tensile stresses in web, lb/in.^{2}

S_{CT} = average compressive axial stress in tension flange of stressed-skin panel due to diagonal tensile stresses in web, lb/in.^{2}

S_{RB} = resultant of the torsional stress S_B and the direct stress S_d experienced by point B in a given weld pattern, lb/in.^{2}

S_S = resultant shear stress in cylindrical tank due to combined loading, lb/in.^{2}

$S_{S(SL)}$ = shear stress in cylindrical tank due to side loading, lb/in.^{2}

$S_{S(TL)}$ = maximum shear stress in cylindrical tank due to weight of tank and contents, lb/in.^{2}

$S_{S(TSL)}$ = torsional stress in cylindrical tank, lb/in.^{2}

$S_{T(A)}$ = tensile stress in cylindrical tank due to acceleration, lb/in.^{2}

$S_{T(max)}$ = maximum tensile stress, lb/in.^{2}

$S_{TB(A)}$ = bending stress in cylindrical tank due to acceleration, lb/in.^{2}

$S_{TB(SL)}$ = bending stress in cylindrical tank due to side loading, lb/in.^{2}

$S_{TB(TL)}$ = maximum bending stress in cylindrical tank due to weight of tank and contents, lb/in.^{2}

S_{TB1} = tensile stress due to primary

LIST OF SYMBOLS (Cont'd)

bending in tensile flange of stressed-skin panel, lb/in.^{2}

S_{TB2} = secondary bending stress in tension flange of stressed-skin panel due to vertical component of diagonal tensile stresses in web, lb/in.^{2}

$S_{T(H)}$ = circumferential tension, or hoop, stress in cylindrical tank, lb/in.^{2}

$S_{T(L)}$ = longitudinal stress in cylindrical tank, lb/in.^{2}

S_a = average dynamic crushing stress of dissipator, lb/in.^{2}

S_b = bearing stress, lb/in.^{2}

S_{b1} = primary bending stress in flanges of stressed-skin panel, lb/in.^{2}

S_c = normal compressive stress, lb/in.^{2}

$S_{c(a)}$ = allowable compressive stress, lb/in.^{2}

$S'_{c(a)}$ = limiting value of allowable compressive stress in stiffener of stressed-skin panel with stiffener on only one side of web, lb/in.^{2}

$S_{c(f)}$ = compressive buckling stress in facing skin of sandwich panel, lb/in.^{2}

$S_{c(max)}$ = maximum compressive stress, lb/in.^{2}

$S_{c(u)}$ = average compressive stress in skin stiffeners (uprights) of stressed-skin panels, lb/in.^{2}

$S'_{c(u)}$ = average compressive stress in stiffener of stressed-skin panel calculated by means of Eq. 4-55 and used in evaluating adequacy of design when stiffener is on one side of web, only, lb/in.^{2}

$S_{c(u)(max)}$ = maximum value of the compressive stress along the length of a stiffener in a stressed-skin panel, lb/in.^{2}

S_{cy} = yield stress in compression, lb/in.^{2}; compressive yield stress at 0.002 in. permanent strain, lb/in.^{2} (used in Eq. 4-52)

S_d = direct stress component (due to direct load component of total load) on an eccentrically loaded welded joint, lb/in.^{2}

S_f = maximum normal stress in facing skin of sandwich panel, lb/in.^{2}

S_i = stress due to moment, or torque, at any point i in an eccentrically loaded weld pattern, lb/in.^{2}

$S_{(max)}$ = general symbol for maximum stress, lb/in.^{2}; maximum resultant stress in cylindrical tank body due to combined loading, lb/in.^{2}

$S_{(min)}$ = minimum spacing between centers of adjacent rivets in same row, in.

S_o = forced crippling stress in stiffeners of stressed-skin panels, lb/in.^{2}

S_p = proportional limit stress, lb/in.^{2}

S_s = total shear stress in web of stressed-skin structures, lb/in.^{2}

S_{s1} = portion of shear stress in web of stressed-skin structure that is resisted by pure shear, lb/in.^{2}

S_{s2} = portion of shear stress in web of stressed-skin structure that is resisted by diagonal tension stresses on appropriate planes, lb/in.^{2}

LIST OF SYMBOLS (Cont'd)

$S_{s(a)}$ = allowable web shear stress in stressed-skin panel, lb/in.^{2}

$S_{s(c)}$ = maximum shear stress in core of sandwich panel, lb/in.^{2}

$S_{s(cr)}$ = critical inelastic (plastic) buckling stress of stressed-skin structure due to shear loading of skin, lb/in.^{2}

$S_{s(cre)}$ = critical elastic buckling stress of skin of stressed-skin structure due to shear loading of skin, lb/in.^{2}

$S_{s(max)}$ = maximum shear stress, lb/in.^{2}

S_{ty} = yield stress in tension, lb/in.^{2}

S_u = ultimate stress at rupture, lb/in.^{2}

S_x = resultant longitudinal stress in cylindrical tank due to combined loading, lb/in.^{2}

S_y = compressive yield stress for column failure, lb/in.^{2}; resultant lateral, or circumferential, tensile stress in cylindrical tank due to combined loading, lb/in.^{2}

T = period of oscillation, sec; torque, lb-in.; ambient air temperature, °F

T_a = ambient air temperature, °C

T_o = turning moment about axis O, ft-lb; toughness index number

T_1 = temperature of heat radiating body, °F abs

T_2 = temperature of heat receiving body, °F abs

t = inside air temperature, °F; time, sec; general symbol for thickness of sheet, plate, sandwich panel, skin of stressed-skin structure, rolled shape, or similar members, in.; total (overall) thickness of sandwich panel, in.; skin thickness in stressed-skin structure, in.

t' = thickness of thickest plate of sheet to be riveted (where t = thickness of thinnest member to be riveted), in.

t_F = thickness of flange in contact with skin of stressed-skin structure, in.

t_c = thickness of sandwich panel core, in.

t_f = thickness of facing skin of sandwich panel, in.

t_o = outside air temperature, °F

t_r = duration of recoil, sec

t_s = thickness of strip material used for core of sandwich panel, in.

t_u = thickness of stiffener in contact with skin of stressed-skin structure, in.

U = overall coefficient of heat transmission, Btu/ht-ft^2-°F

U_r = modulus of resilience

U_t = modulus of toughness

V = volume, in.^{3} (also ft^3); vertical shear load acting on web of stressed-skin structure, lb; vertical shear load on beam of unit width, lb/in.; relative flow velocity, fps

$\overline{V}$ = displaced volume of water, ft^3

V_1, V_2 = submerged volumes 1 and 2, respectively, ft^3

V_H = heaped capacity, ft^3 (also yd^3)

$V_{(max)}$ = maximum vertical shear on unit width of sandwich panel loaded as beam, lb/in. of width; maximum vertical shear load, lb

V_p = velocity of projectile, fps

V_r = velocity of recoiling parts, fps

v = velocity, fps

v_x = velocity in x direction, fps

v_{x1} = initial velocity in x direction, fps

LIST OF SYMBOLS (Cont'd)

v_{x_2} = final velocity in x direction, fps
v_y = velocity in y direction, fps
v_{y_1} = initial velocity in y direction, fps
v_{y_2} = final velocity in y direction, fps
W = general symbol for weight, lb; static weight of a falling load, lb; weight of explosive charge, lb; wind velocity, mph
W_p = total payload of heaped cargo box, lb
W_V = weight of vehicle, lb
W_v = wind velocity, m/sec
w = weight of component, lb; inside width of cargo box, in.
x = linear displacement in direction indicated as x direction, ft; thickness of material parallel to heat flow, ft
$x_1, x_2, x_3, \ldots, x_n$ = respective thicknesses of layers 1, 2, 3, ..., n, ft
y = linear displacement in direction indicated as y, ft; amplitude of vibration, in.; perpendicular distance from centroidal axis of section to outermost fiber (also to any fiber), in.
$\dot{y}$ = velocity, in./sec
$\ddot{y}$ = acceleration, in./sec^2
y' = distance from neutral axis of flange section of stressed-skin panel to flange fiber under consideration, in.
Z = section modulus, in.^{3}
Z'' = depth to point on amphibious vehicle that first contacts bank prior to emerging from water, ft

Greek Letters

a = angle of diagonal tension in stressed-skin panel measured from longitudinal neutral axis, deg
β = angle of trim, deg
Δt = duration of impact, sec; temperature difference between fluid and surface, °F
$\Delta_x t$ = temperature difference across thickness of material, °F
δ = deflection, in.
δ' = beam deflection under dynamic loading, in.
$\delta_{(max)}$ = maximum deflection of panel or beam, in.
$\delta_{(max)_a}$ = maximum deflection with loading per condition a, in.
$\delta_{(max)_b}$ = maximum deflection with loading per condition b, in.
ϵ = strain
ϵ_r = ultimate strain at rupture
θ = thread half-angle, deg
μ = Poisson's ratio, dimensionless
μ_B = friction coefficient of nut or bolt bearing surface, dimensionless
μ_T = friction coefficient at thread contact surfaces of threaded fastener, dimensionless
ρ = general symbol for density, lb/ft^3 (also lb/in.^{3} and slug/ft^3); mass density of air, slug/ft^3 (=0.00237 slug/ft^3 at 60°F and standard atmospheric pressure)
γ = density of air, lb/ft^3 (=0.0764 lb/ft^3 at 60°F and standard atmospheric pressure); angle of roll, rad (also deg)

LIST OF SYMBOLS (Concluded)

γ_o = angular displacement when time $t = 0$, rad

$\dot{\gamma}_o$ = angular velocity when time $t = 0$, rad/sec

ϕ = thread helix angle, deg

ω = angular velocity, rad/sec

ω_1 = initial angular velocity, rad/sec

ω_2 = final angular velocity, rad/sec

PREFACE

This handbook is one of the Automotive Series of the Army Materiel Command's Engineering Design Handbooks. This is a coordinated series of handbooks that contain basic information and fundamental data useful in the design and development of Army materiel and systems. They are authoritative references of practical information and quantitative data that serve as guidelines in the development of materiel to fulfill the tactical and technical requirements of the Armed Forces. Their purpose is to provide assistance to designers of military equipment and a measure of specialized technical guidance to military and civilian personnel who are responsible for the preparation of specifications for this equipment, and to those who are responsible for ensuring that these specifications are fulfilled.

The Automotive Series treats the broad field of automotive equipment design, with particular emphasis upon the physical, operational, climatic, and human factors considerations imposed by the military environment and upon the ways that these considerations are unique to that environment.

The vehicle body or hull, one of the major functional elements of the automotive assembly, is the subject of this handbook. Its treatment is divided into four major chapters; namely, Chapter 1, "Introduction", Chapter 2, "Design Considerations", Chapter 3, "Components Associated With Body and Hull Design", and Chapter 4, "Design Procedures and Design Criteria". A glossary of special terms used in this handbook and an index are included at the end of the handbook.

The material presented in this handbook is a compilation of data and design information gathered from many sources. Because of the vast scope of the subject, its treatment has been limited to one of condensation and summary. Proofs and mathematical derivations are omitted, for the most part. Only final equations, qualifying or limiting information, and interpretations useful to the design process are given. For a comprehensive treatment of any topic discussed in this handbook, the reader is urged to consult the pertinent references listed at the end of each chapter.

When reference is made in this handbook to specific military or civilian specifications, regulations, or other official directives, it is done so to inform the reader of the existence of these documents. In this respect, the user is cautioned to make certain he uses editions that are current at the time of their use.

This handbook was prepared by the Military Systems and Automatic Equipment Section, Engineering Mechanics Division*, of the IIT Research Institute, Chicago, under subcontract to the Engineering Handbook Office of Duke University, prime contractor to the U. S. Army for the Engineering Design Handbook Series. Technical guidance and general assistance were provided by the U. S. Army Tank-Automotive Command, Warren, Michigan.

The Handbooks are readily available to all elements of AMC including personnel and contractors having a need and/or requirement. The Army Materiel Command policy is to release these Engineering Design Handbooks to other DOD activities and their contractors, and other Government agencies in accordance with current Army Regulation 70-31, dated 9 September 1966. Procedures for acquiring these Handbooks follow:

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*Rudolph J. Zastera, Project Leader and Principal Author

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Department of the Army
Washington, D. C. 20310

All requests, other than those originating within the DOD, must be accompanied by a valid justification.

Comments and suggestions on this handbook are welcome and should be addressed to Army Research Office-Durham, Box CM, Duke Station, Durham, N. C. 27706.

CHAPTER 1
INTRODUCTION*

SECTION I—GENERAL

The art of designing is a complex combination of imagination, creative ability, ingenuity, knowledge of scientific principles and manufacturing techniques, a familiarity with materials, skill, and personal zeal. This handbook does not presume, nor attempt, to teach this art. Quite to the contrary, this handbook is prepared on the presumption that it will be used by engineers and designers who have attained a level of education equivalent to a B. S. degree in engineering and who have had some experience in the design of mechanisms or structures but not specifically related to the military automotive field. Its purpose is to supplement the knowledge of these engineers with a ready reference of design guidelines, objectives, and pertinent specifications and standards based upon field experience, regulations and directives, and the desires of the field forces who are the ultimate users of the equipment.

The first handbook (AMCP 706-355, *The Automotive Assembly)* of the Automotive Series of Engineering Design Handbooks represents the automotive assembly as comprising several principal elements or systems such as the power plant, power train, suspension system, electrical system, and others. Since each of these elements is quite complex, a detailed discussion of the principles and procedures governing their design is relegated to separate handbooks, each dealing with one particular element. In keeping with this plan, this handbook is devoted to a discussion of body and hull design for military land vehicles of all types. Its principal objectives are (1) to orient design personnel in the requirements prescribed for the bodies and hulls of military land vehicles, (2) to discuss the various considerations, problem areas, and procedures involved in effecting a body design that will meet these requirements, and (3) to present design criteria in the form of formulas, charts, and tables of quantitative data useful in the design of military automotive bodies.

It should be noted that this handbook is primarily concerned with military vehicles. As such, matters of body styling for purely aesthetic reasons, and considerations of comfort and convenience beyond those necessary for the effective operation of the vehicle and the accomplishment of its mission, have no place in it. Furthermore, the subject of body and hull design for all types of military vehicles comprises such a vast number of topics and subtopics that a detailed, comprehensive coverage in one convenient, useful volume is an impossibility. In order to reduce this task to one of more practical proportions, a generalized approach is used, wherever this is suitable, in order to make the discussions applicable to a large number of similar cases. Users of this handbook should not expect to find complete, step-by-step procedures on every conceivable body design problem they might ever encounter. Where procedures are given, they apply to examples considered to be typical or particularly illustrative. The assumption is made that the users of this handbook are competent to modify the procedures given when necessary to make them applicable to specific cases.

The same assumption applies with respect to the units assigned to mathematical symbols in this book. Unless otherwise noted, the system of units used for force, length, and time is the British gravitational system of lb, ft, and sec, respectively. Due care should be observed, when converting to another system of units, to change the values of any numerical constants appearing in the equations in conformance with the new system.

1-1 TERMINOLOGY

Many of the terms used in this handbook have precise connotations when applied to military vehicles. Even the term "military vehicle," as used here, connotes more than merely a vehicle that is used by the Army; nor does it necessarily imply strictly offensive types, or fighting,

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vehicles—although these are included in the complete definition of the term. Before proceeding with this discussion, it is important to have a clear and complete understanding of the terminology used. Some of the fundamental terms are discussed in the paragraphs which follow. Other terms that have somewhat specific definitions when applied to the subject of military automotive vehicle bodies, or hulls, are listed in the Glossary at the end of this handbook. Definitions of other military terms can be found in Refs. 1 and 2.

1-1.1 MILITARY VEHICLE

The meaning of the term "military vehicle", as it is used in this handbook, is restricted to vehicles that travel on land as opposed to aircraft of all types. Traveling on land, however, embraces the ability to negotiate water barriers by fording or swimming; hence, amphibious vehicles of all types are included in the term. Military vehicles comprise all forms of wheeled and track-laying vehicles equipped with the full gamut of body types found in commercial vehicles plus special bodies and equipment that are unique to military operations. They include all types of trucks, tractors, truck-tractors, personnel carriers, tanks, self-propelled guns, motorized and mechanized special purpose equipment, trailers, vans, sleds, and special-purpose towed vehicles.

The principal distinction between these vehicles and their commercial counterparts is that military vehicles are *specifically* designed for military purposes. This includes combat operations and the transportation of cargo, personnel, or equipment—or for towing other vehicles or equipment—cross-country and over roads in close support of combat vehicles and troops. They are designed and constructed to endure the rigors of the military environment and to continue to operate at, or above, a prescribed minimum performance level in this environment. They have excellent cross-country performance capabilities over all types of terrain where tactical or combat operations can be conducted. This includes snow and ice, muskeg and swamp, rocky terrain, and desert sands. In order to negotiate water barriers with a minimum preparation, all sensitive equipment is either permanently waterproof to prevent damage due to immersion, or else is designed to function underwater. The current trend is toward making all military vehicles amphibious, i.e., give them the capability of floating or swimming. This permits them to cross water barriers of any depth. Some amphibious vehicles can even operate in fairly heavy surf conditions, although at the present time, most find it difficult to breast a current very much greater than 3½ knots.

Some other features and characteristics that are standard on military vehicles are: 24 volt, fully water-proofed, fungus-proofed electrical systems fully suppressed to prevent interference to electronic equipment; engines that are capable of operating while fully submerged in either fresh or sea water; oversized air, oil, and fuel filtering capacities; oversized generator capacity; oversized oil and engine cooling capacities; engines that are less critical of the fuels they require and have a lower specific weight (lb per bhp); reliability over an extremely wide temperature range; provisions for operating during blackout conditions; all-wheel drive on wheeled vehicles; and improved ease of servicing and maintenance.

1-1.2 COMBAT VS TACTICAL VEHICLES

Military vehicles are classified into three broad categories; namely, combat vehicles, tactical vehicles, and administrative vehicles. The latter category, administrative vehicles, comprises the standard, commercially available vehicles used largely at camps, posts, stations, and various Government installations for routine administrative duties. Vehicles of this category are of only minor concern to the military designer. Combat vehicles and tactical vehicles, however, are his total concern; therefore, some time should be spent on their distinction.

1-1.2.1 Combat Vehicles

Combat vehicles are defined[1*] as land or amphibious vehicles, with or without armor or armament, designed for specific functions in combat or battle. The later installation of armor or armament onto other than combat vehicles does not alter their original classification. They may be wheeled or track-laying; but in all cases, they are designed to have a high degree of

*Superscript numbers refer to References at the end of each chapter.

mobility in off-road operations. Some typical combat vehicles are tanks, self-propelled artillery, missile launchers, and armored cars.

The majority of combat vehicles at the present time are track-laying vehicles, but this is not a requirement of this classification. Research in the field of land locomotion has shown that equal mobility can be obtained with wheels as with tracks in identical terrain and with equal vehicle loads if the full potential of the wheels is exploited[3-5]. Furthermore, due to their missions, combat vehicles are usually equipped with both armor and armament; although certain antitank vehicles are unarmored and depend upon their resulting decreased silhouette and increased speed and maneuverability for protection. A continuing demand for increased firepower and increased mobility has resulted in an increasing use of lightweight armor on vehicle bodies and hulls. The resulting reduction in vehicle weight enables what would otherwise be heavy vehicles to partake in airborne operations and it also improves their amphibious capabilities. Weight reduction in the newer types of combat vehicles is largely due to the extensive application of aluminum in their construction, including aluminum armor.

1-1.2.2 Tactical Vehicles

Tactical vehicles are generally defined as vehicles that have been designed and manufactured specifically to meet the severe requirements imposed by combat and tactical operations in the field. Whereas combat vehicles are defined (par. 1-1.2.1) as vehicles designed to perform specific functions in combat, tactical vehicles are vehicles specifically designed to support the tactical play of the operation. Military tactics is that branch of the military art that deals with the arranging, positioning, and maneuvering of the forces in or near contact with the enemy, and the maneuvering and positioning of materiel and supplies in support of the forces in contact, so as to attain an objective in a campaign or battle, to achieve some immediate advantage, or to ameliorate a disadvantage[1]. Since the main purpose of tactical vehicles is to give direct support to the combat vehicles, they are required to have the same high level of mobility as do combat vehicles; and like combat vehicles, they are designed to exacting military characteristics to survive and perform satisfactorily in the military environment.

1-1.3 BODIES VS HULLS

1-1.3.1 General Discussion

The terms *body* and *hull* are used somewhat interchangeably in military automotive circles to designate the portion of an automotive vehicle that comprises the functional compartments, i.e., the compartments which house the cargo, passengers, or crew.

As originally used, the term hull was restricted to the enclosing structures of track-laying tank-type vehicles, while body was used with reference to wheeled vehicles. However, the bodies of early wheeled amphibious vehicles functionally and structurally resembled the hulls of boats or barges and, as a result, were referred to as hulls rather than as bodies. Many contemporary designs of these vehicles no longer have this boat-like appearance, but common usage has retained the term hull rather than body for their enclosing structures.

Generally speaking, then, track-laying and amphibious vehicles have *hulls*, while nonamphibious wheeled vehicles have *bodies.* Exceptions to this general statement are certain high speed track-laying tractors used as prime movers for heavy artillery, such as the M6 Tractor, and certain types of high-speed, heavy-duty, track-laying cargo tractors, such as the M85 (Fig. 1-27). The enclosing structures of these nonamphibious, unarmored vehicles resemble the bodies of their wheeled counterparts to such a degree that they are often referred to as bodies rather than hulls. Strictly speaking, however, the term hull is considered correct for these vehicles.

1-1.3.2 Hulls

The hull of a tracked vehicle—such as a tank, armored personnel carrier, or armored car—is the massive enclosing structure that gives the vehicle its characteristic appearance exclusive of the tracks and suspension system, the propulsion system, the turret and cupola assemblies, and the vehicle's armament. It may be fabricated as a one-piece welded assembly of armor plate (ferrous or nonferrous), as a one-piece armor

casting, or as a combination of plates and castings. Certain lightweight hulls consist of a metal framework covered with plates, while others use plates of sufficient thickness and rigidity that a load carrying frame is not necessary. In any case, the completed hull serves as the main frame, compartment and housing for the propulsion system, crew, passengers, cargo, and equipment, and provides a base for mounting the turret (in the case of tank-type vehicles), the suspension components, and the armament. Hinged or removable panels are provided for personnel access to the interior, for the installation, servicing, and removal of components, and for the loading and unloading of ammunition and cargo. In addition, provisions are made on the hull for towing, hoisting, and tiedown of the completed vehicle. Vision is provided for by installing lights, windows, ports, vision blocks, and periscopes. All of these features, the problems they present, and the design considerations they require are discussed in some detail in other portions of this book. A typical tank hull, without any attachments, is shown in Fig. 1-1. The complete vehicle having this hull is shown in Fig. 1-23.

Hull-type wheeled vehicles are generally amphibious vehicles, as is explained in par. 1-1.3.1; the exceptions are the armored cars (Fig. 1-2). These, in the past, have been much too heavy to float and, therefore, were not amphibians. With modern lightweight construction techniques and lightweight armor materials, it is conceivable that amphibious armored cars could be made a reality if there were a requirement for such a vehicle. Nevertheless, the body of the nonamphibious wheeled armored car is considered to be a hull, perhaps due to its appearance, and is, therefore, an exception to the rule.

The design and general appearance of wheeled amphibious vehicles divide them into two categories: those that resemble land vehicles, of the type illustrated in Figs. 1-7 and 1-8; and those resembling boats or barges, of the type illustrated in Figs. 1-17 and 1-18. In both categories, the hull is defined as the entire structure that encloses the crew, passengers, cargo, propulsion system, and equipment. Functionally, wheeled vehicle hulls are identical to tracked vehicle hulls and differ from them only in physical appearance. The wheeled vehicle hull must be watertight and must provide sufficient buoyancy to float the vehicle with proper trim and stability for safety. Vehicle hulls of the type shown in Fig. 1-18 are designed in accordance with the best principles of marine engineering. The wheels are fully retractable into covered wheel wells to minimize drag resistance during water operations.

1-1.3.3 Bodies

The body of a wheeled vehicle is that assembly of panel sections, framing, doors, and windows that form the main enclosure for cargo,

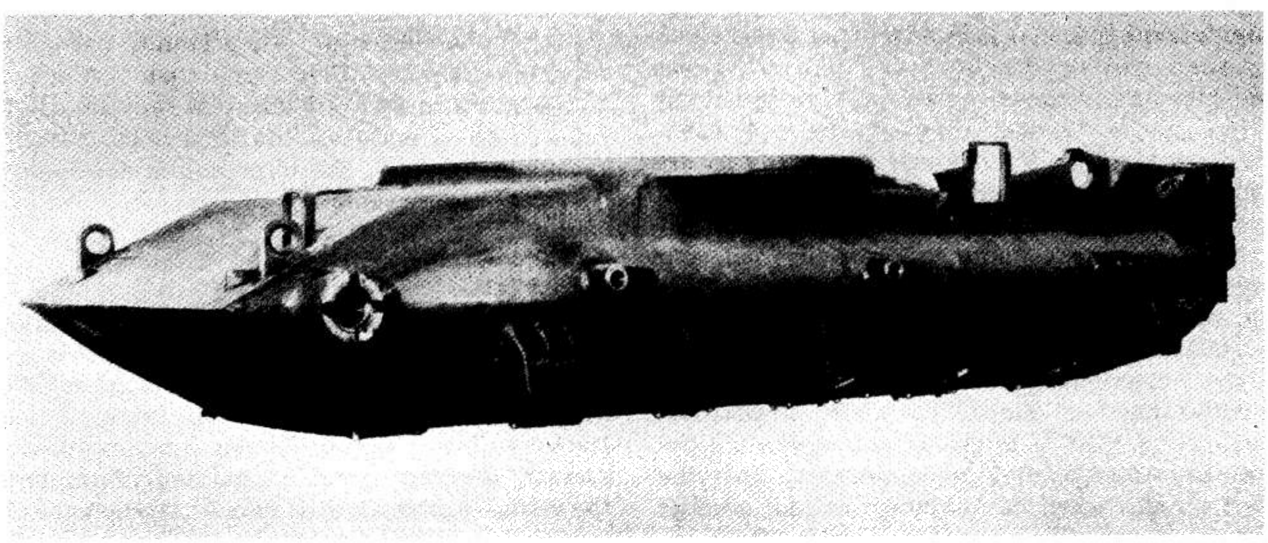


Figure 1-1. Typical Tank Hull

Figure 1-2. Armored Car, M8E1—1945

passengers, and/or equipment in accordance with the functional purpose of the vehicle. It excludes the chassis assembly and, in all but a few special cases, also excludes the cab. The special cases are those vehicle types which do not have a separate cab, the crew compartment being part of the body. Examples of vehicles of this type are sedan cars, busses, some ambulances, and the 1/4-ton utility vehicles.

A large number of body types exist, each designed for some functional purpose, such as hauling bulk loaded cargo, hauling and dumping construction materials, hauling liquids or gases, transporting personnel, or transporting special equipment. These are discussed in greater detail in subsequent sections of this chapter. A typical characteristic of many military vehicle bodies is their flexibility to be used for hauling more than one general type of cargo. Thus, a dump truck, intended primarily for the hauling and placing of bulk cargo, when provided with bows and a tarpaulin cover becomes an effective transport for general cargo and, with the addition of troop seats, for personnel.

The current trend in military wheeled vehicle design is for all wheeled vehicles to have amphibious capabilities, i.e., to be able to cross inland bodies of water by swimming rather than by using fording techniques. This requirement had led to the adoption of lightweight construction techniques, an increase in the ratio of payload weight to vehicle weight, the provision of flotation chambers, and greater concern over weight distribution.

1-1.3.4 Trailer Bodies and Hulls

Trailers, vans, and special-purpose towed vehicles, although not self-propelled and, therefore, not automotive vehicles in the strictest sense of the definition, are also included in the scope of this handbook. In general, towed vehicles must meet the same rigorous specifications as the vehicles that provide their motive power, and present the same design problems as do the prime movers. When trailers are designed to accompany amphibious vehicles, the trailer must also meet all required amphibious requirements. This means that the hull must be sufficiently buoyant to support its loaded weight with sufficient freeboard for safety and must follow behind the towing vehicle in the water without any unacceptable tendency to yaw, fishtail, or porpoise.

Both wheeled and track-laying trailers are in existence, although track-laying trailers are not as common. Trailer bodies are made in all of the types that are available for self-propelled

vehicles; and design considerations, procedures, requirements, and fabrication techniques are basically the same as for their powered counterparts. These topics are discussed in other portions of this handbook.

1-2 THE MILITARY ENVIRONMENT

A first and continued consideration of any designer is the operating environment of the product of his design activities. A mechanism or device to be used in the environs of a hospital operating room will have markedly different characteristics than a similar device designed for use in an oilfield or a coal mine. Equipment and hardware intended for aerospace application are characteristically different from railroad equipment. Similarly, military equipment should be expected to differ from its counterparts in the civilian field. As the natural environment plays a major role in shaping and developing the plants and animals of nature, so does the military environment play a major role in the shaping and developing of military vehicles and equipment. It is important, therefore, that the designer of military vehicles be thoroughly acquainted and impressed with the nature of this environment.

The environment in which military vehicles are required to operate is the most severe found anywhere on earth and has no civilian counterpart. This is a difficult concept for inexperienced engineers and designers to accept and fully appreciate but one with which the experienced military engineer will agree only too readily. And despite the severity of the military environment, the fallacious procedure of overdesigning to meet this severity brings unacceptable penalties in the form of increased weights, increased power requirements, a greater vehicle silhouette, increased fuel requirements, decreased mobility, and a less efficient vehicle at an increased cost.

The environment of the military vehicle is the battlefield and the terrain leading to and away from the battlefield. It includes forests and fields, swamps and deserts, mountains and plains, mud, sand, dust, rain, snow, and ice. It includes airborne and airdrop operations, amphibious and deep-fording operations, cross-country (off-road) operations in all types of terrain, and a multitude of loading, unloading, and shipping operations. It is often an environment of high shock, bombardment, fire, frantic movements, and chaos conducted in all extremes of weather and climates ranging from intense tropic heat to frigid arctic cold. Maintenance is sometimes neglected, operators' skills are sometimes wanting, and vehicle abuse through overload and hard service is commonplace.

Tactics envisioned by military planners for future wars place maximum emphasis on rapid cross-country mobility in all types of terrain, weather, and climate. This necessitates that military vehicles be designed specifically for this specialized purpose, and no amount of adaptation or modification of commercial vehicles can produce an entirely satisfactory result. Countless efforts in this direction in the past have proven this beyond any reasonable argument.

As tactical thinking is directed more and more toward chemical, bacteriological, and radiological warfare, the military environment takes on new dimensions of severity. To meet the new problems introduced by this type of warfare, additional protection for personnel and sensitive cargo and equipment is required. This may be accomplished by designing the personnel compartments to give greater protection; sealing the compartments against the entrance of contaminants through doors, windows, access openings, structural joints, and ventilating systems; incorporating closed ventilating systems using rejuvenated and recirculated air; and providing for all personal requirements of the crew and passengers during extended periods of time. Sealing of personnel compartments and radiation protection increases the problems of vision out of the vehicle and the problems of serving the major vehicle-mounted weapons. The former necessitates the use of periscope vision devices and the latter often requires automatic loading devices and remote aiming and firing mechanisms.

Sealing and insulating crew compartments against chemical, bacteriological, and nuclear agents also effectively seal out sound, including battle sounds. Work done with certain experimental vehicles brought to realization the fact that a consciousness of the sounds about them is necessary to the efficient performance of combat crews. As a result, the installation of auxiliary equipment was needed to restore sound orientation.

Other factors that are unique to the military environment are the need for protection against attack by conventional weapons and requirements resulting from the long battle day specified for military vehicles. As applied to combat tanks, the battle day specified is 24 hours long; while with respect to armored personnel carriers, the battle day is specified as 3 days long. The significance of this is that these vehicles must be able to conduct typical operations for these specified periods without refueling or maintenance and provide for all requirements of the crew. The application of armor to a vehicle brings additional problems of weight, size, power requirements, ventilation, and vision. Battle-day length has a direct influence upon required fuel capacity, maintenance requirements, ammunition stowage, and crew requirements.

A considerable amount of information exists in the open literature and in various military publications [6-8] on the nature of the military environment and its impact upon the design of military equipment. The military designer should become thoroughly familiar with these concepts and endeavor to design equipment that will be compatible to this environment. The salient features of the military environment are summarized in the following list:

a. High shock and vibration produced by off-the-road operation over rough terrain, airdrop operations, high explosive blast, and ballistic impact.

b. Extreme temperature ranges, extending from arctic to tropical.

c. Operations under conditions of extreme dust.

d. Operations in deep mud.

e. Operations in snow and on ice.

f. Amphibious operations in both fresh and sea water.

g. Operations under conditions conducive to corrosion and fungus growth.

h. Mountain operations involving extremely long, steep grades and side slopes.

i. Extended operations under conditions of low speed and high load.

j. Operator abuse in the form of overload, misuse, improper maintenance, and neglect.

SECTION II—TYPES OF WHEELED VEHICLES

1-3 GENERAL DESCRIPTION

The majority of wheeled vehicles fall into the tactical rather than the combat vehicle category (pars. 1-1.2.1 and 1-1.2.2), although wheeled combat vehicles do exist. The presence of wheels vs tracks, however, is not a determinant in vehicle classification. At present, tactical vehicles that accompany troops into actual contact with the enemy, or that follow in immediate support of the combat units, are generally of the track-laying type; although wheeled vehicles are also used in this role but to a lesser degree. Tactical vehicles that are in follow-up roles—such as general resupply, construction, casualty evacuation, equipment recovery, maintenance and service, and command and communication—are generally wheeled vehicles, although again, there are exceptions to this.

Wheels perform better at high speeds than do tracks, particularly over relatively smooth terrain. They are more quiet, have a longer service life, are easier to steer, and are generally less expensive than track assemblies. Tracks, on the other hand, are less vulnerable to the impact of projectiles than are pneumatic tires; they make it easier to attain a low ground pressure under vehicles, where this is desirable, particularly true in the case of very heavy (50 tons gross weight and over) vehicles; and they usually permit a lower vehicle silhouette than is obtainable in a wheeled vehicle of comparable weight and ground pressure.

The popular belief that tracks are superior to wheels for cross-country operations, however, is a fallacy. It has been shown [3-5] that equal mobility can be attained with wheels as with tracks in identical terrain and under identical loads if soil-vehicle relationships are correctly utilized during the vehicle design phase. Other considerations cause preference to be given to one suspension type over the other.

The type classification of wheeled vehicles is based upon their primary function and, in some instances, upon distinguishing characteristics in their design. The fundamental function of all vehicles, whether combat or tactical, is to transport some type of cargo. This cargo may be personnel, bulk cargo, construction material,

weapon systems, special-purpose equipment, or another vehicle as is the case with prime movers or retrieving types of vehicles. In order to reduce the total number of vehicles required by the Armed Forces, military vehicles are designed to be capable of performing secondary functions in addition to their primary ones. Thus, a vehicle designed primarily to be a carrier of bulk cargo is provided with removable or foldaway seats to enable it to transport personnel in relative comfort, and with a pintle assembly so that it can also be used as a prime mover for towing trailers, artillery, or other equipment.

A general knowledge of the various vehicle types necessary to the conduct of modern military operations is useful background to the designer of military automotive equipment. Toward this end, brief descriptions of some representative wheeled vehicle types are given in the paragraphs which follow. Representative types of tracked vehicles are discussed in Section III of this chapter. A comprehensive inventory of all military vehicles, including illustrations and specifications of each, are given in Refs. 9, 10, and 11.

1-3.1 WHEELED COMBAT VEHICLES

In the combat vehicle category (par. 1-1.2.1), wheeled vehicles have only a meager representation at present because the predominant development of vehicles in this category have been track-laying types. A typical example of a wheeled combat vehicle is the armored car shown in Fig. 1-2. This vehicle is obsolete at present but renewed interest is being shown in vehicles of this type. It is both armed and armored, and is equipped with radio communication equipment. Its primary function, in addition to providing high-speed mobility, is to perform combat reconnaissance and provide defense firepower when needed. Vehicles of this type have also been used as field commanders' vehicles in mechanized and armored units. They have secondary functions as personnel and cargo carriers. Some models existed whose primary function was to provide mobile armored protection for personnel and cargo. In these vehicles, the turret was omitted and the hull was modified to facilitate cargo loading and unloading. By applying lightweight construction techniques and lightweight armor, vehicles of this type can be made amphibious and readily air transportable.

1-3.2 WHEELED TACTICAL VEHICLES

Wheeled vehicles, as has been stated earlier in this chapter, generally belong to the tactical vehicle category (par. 1-1.2.2); although they do not constitute all of this group. Many tactical vehicles are track-laying types. The major types of wheeled tactical vehicles, classified by their primary functions, are:

a. Passenger vehicles
b. Utility platform trucks
c. Utility trucks
d. Cargo trucks
e. Dump trucks
f. Cargo dump trucks
g. Tank Trucks
h. Van trucks
i. Maintenance trucks
j. Wrecker trucks
k. Truck tractors
l. Tractor wrecker trucks
m. Trailers, semitrailers, and tank transporters

Passenger vehicles include busses, sedan cars, and ambulances. Their primary function is to transport personnel. Busses, sedan cars, and certain types of ambulances (types that are similar to civilian ambulances) fall into the administrative category of vehicles (par. 1-1.2) and, as such, are of little interest to the military vehicle designer since they are purchased directly from the automotive industry. Frontline and field ambulances, of the type shown in Fig. 1-3, are tactical vehicles in every sense of the word and are designed to rigorous military specifications. Their function is to transport sick and wounded personnel either seated or on litters. The patients' compartment is often provided with personnel heaters, a surgical light, and ventilating equipment.

Utility platform trucks are vehicles of the type illustrated in Fig. 1-4. They are essentially platforms mounted on two axles and four wheels and are powered by an engine mounted under the platform. Their function is to provide a means of transporting light cargo over rough terrain at slow speeds, keeping pace with riflemen moving at foot speeds in combat.

Utility trucks are the short wheelbase, light-load capacity (1/4-ton) vehicles popularly

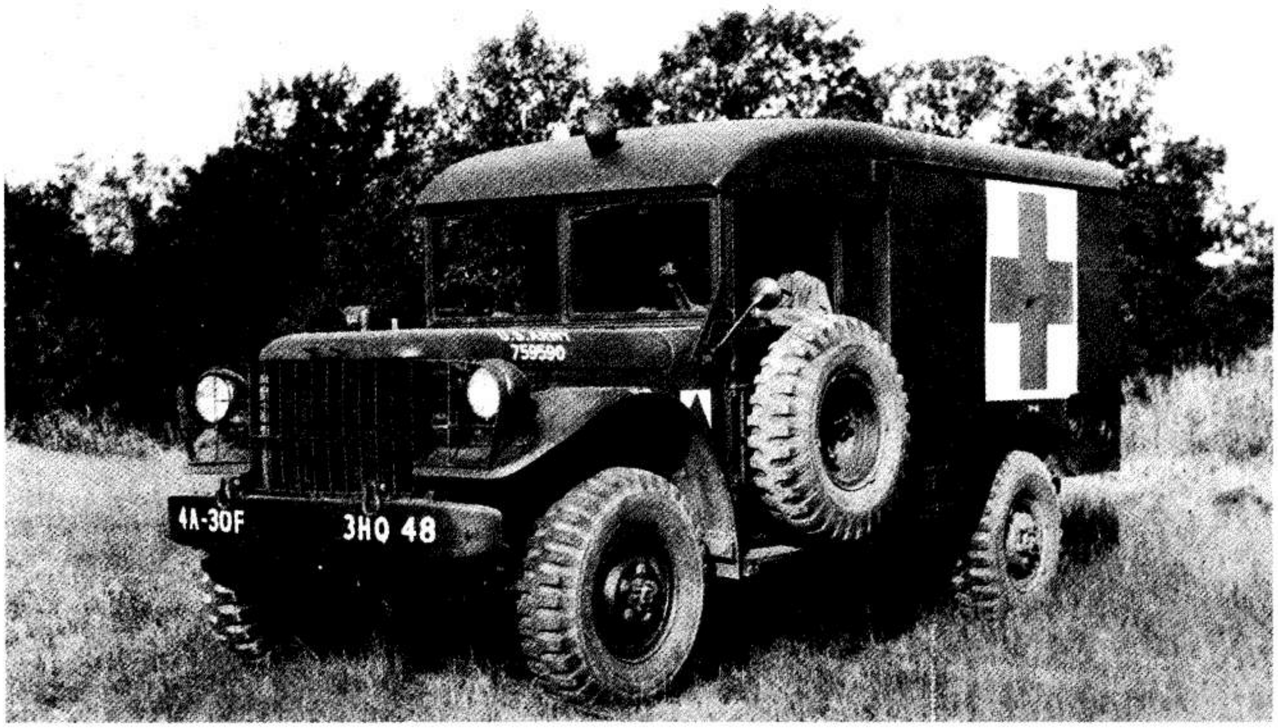


Figure 1-3. Ambulance Truck, 3/4-ton, 4 X 4, M43

known as "Jeeps" (Fig. 1-5). These vehicles are designed for service as general purpose personnel or cargo carriers and are especially useful as command, reconnaissance, and communications vehicles and for mounting recoilless weapons.

A military *cargo truck* (Fig. 1-6) is a standard military automotive chassis upon which is installed a typical military cargo body. The latter consists of a box-type metal bed which forms the floor, sides, and front end, a hinged tailgate, lattice type side extensions (usually removable), and removable top bows and tarpaulin cover. It is designed to transport general cargo and personnel. Removable troop seats and lazy backs, usually of wood, provide a degree of comfort for personnel.

Two newer versions of military cargo trucks are shown in Figs. 1-7 and 1-8. Both of these vehicles are amphibious. Their hulls are completely watertight and sufficiently buoyant to enable the loaded vehicle to float with reasonable freeboard. Propulsion and steering in the water is provided by the wheels. Neither of these vehicles is built upon a standard automotive chassis. The M656 shown in Fig. 1-7 has an integrated frame and body, takes full advantage of lightweight materials and lightweight construction techniques, and is a vehicle that was designed completely as a special vehicle; i.e., it is not a modification or adaptation of a commercial vehicle.

The XM520 Cargo Truck, shown in Fig. 1-8, is referred to as an exoskeletal type of construction in that its frame members are on the outside of the vehicle. It is patterned after certain characteristic types of very large wheeled earthmoving machines that have this general configuration. Vehicles of this type, featuring extremely large diameter wheels (29.5 X 25 tires), exoskeletal construction, and wagon steering, are commonly known as "Goer" vehicles; and this general vehicle concept is known as the "Goer concept".

Both of these vehicle types have demonstrated excellent performance characteristics and may well be the pattern that future military vehicles will follow. Both concepts are adaptable to other hull types.

Dump trucks and *cargo dump trucks* (Fig. 1-9) are similar to cargo trucks except that the body is mounted upon a transverse hinge and a powered hoisting mechanism is added to tilt the

Figure 1-4. Utility Platform Truck, 1/2-ton, 4 X 4, M274

cargo body about the hinge, thus unloading the contents. These vehicles are designed to transport and dump such materials as earth, sand, gravel, coal, etc. In addition, they can also be used for transporting general cargo and personnel.

Tank trucks (Fig. 1-10) are standard military chassis equipped with tank bodies designed for transporting water, liquid fuels, oils, liquid gases, and other fluids. Equipment for dispensing and filtering the fluids is usually included as part of the vehicle.

Van trucks (Fig. 1-11) are automotive vehicles that have fully enclosed box-like bodies, not integral with the cab, that are designed to protect their cargo and equipment from the elements and pilferage and, in some cases, to provide working quarters for personnel. Various types of vans exist to satisfy various needs. Some have windows, some have expansible sides, some are insulated, and some have internal lighting and ventilating or air conditioning systems. They are used to house and transport maintenance and repair shops; electronic communication, surveillance, test, and control equipment; map-making and reproduction facilities; and as mobile offices and quarters for general officers.

A *maintenance truck* is one that is equipped with a body specifically designed to house and transport tools, equipment, and supplies necessary for the efficient performance of some specific type of maintenance, repairs, or construction in the field. For example: a Signal Corps maintenance truck would be designed and equipped for telephone and electric line construction, setting poles, pulling wire, and telephone repair work. It might also include a

Figure 1-5. Utility Truck, 1/4-ton, 4 X4, M151

powered earth auger, a crane, and cradles for wire or cable reels. Similar trucks are designed for ordnance maintenance, vehicle maintenance, and other special fields.

Wrecker trucks (Fig. 1-12) are designed to tow, salvage, and recover disabled vehicles and other mechanized equipment and to perform various emergency repair operations on vehicles. They consist of a standard automotive chassis upon which is mounted a platform body, usually surfaced with safety-tread plate, and supporting a hydraulic crane. Four outriggers, two on each side, are usually attached to the platform frame to remove the load from the vehicle's suspension system and tires during heavy lifting operations. Sockets are provided along the sides of the platform for installing top bows to support a tarpaulin with end curtains for camouflage purposes, when necessary. The load rating of wreckers—e.g., 2-1/2 ton, 6 X 6—refers to the chassis and not to the lifting capacity of the crane. The maximum rated lifting capacity of the 2- 1/2-ton, M60 Wrecker shown in Fig. 1-12 is 8,000 lb. A comparable 5-ton wrecker, M62, has a crane with a maximum lifting capacity of 20,000 lb.

Truck tractors (Figs. 1-13 and 1-14) are automotive wheeled vehicles designed to tow and partially support a semitrailer by means of a fifth wheel-type coupler. In addition, they supply all electrical and braking requirements of the semitrailer. Some truck tractors, particularly those of large capacity, are equipped with powered winches mounted aft of the cab. These are used to pull disabled vehicles onto coupled semitrailers or tank transporters. Some models (Fig. 1-14) are equipped with vertical lifting devices, used in conjunction with the winch, which enable them to operate as recovery vehicles independently of the semitrailer.

Tractor wrecker trucks are essentially truck tractors equipped with a hydraulically operated extendible boom mounted in the rear of the

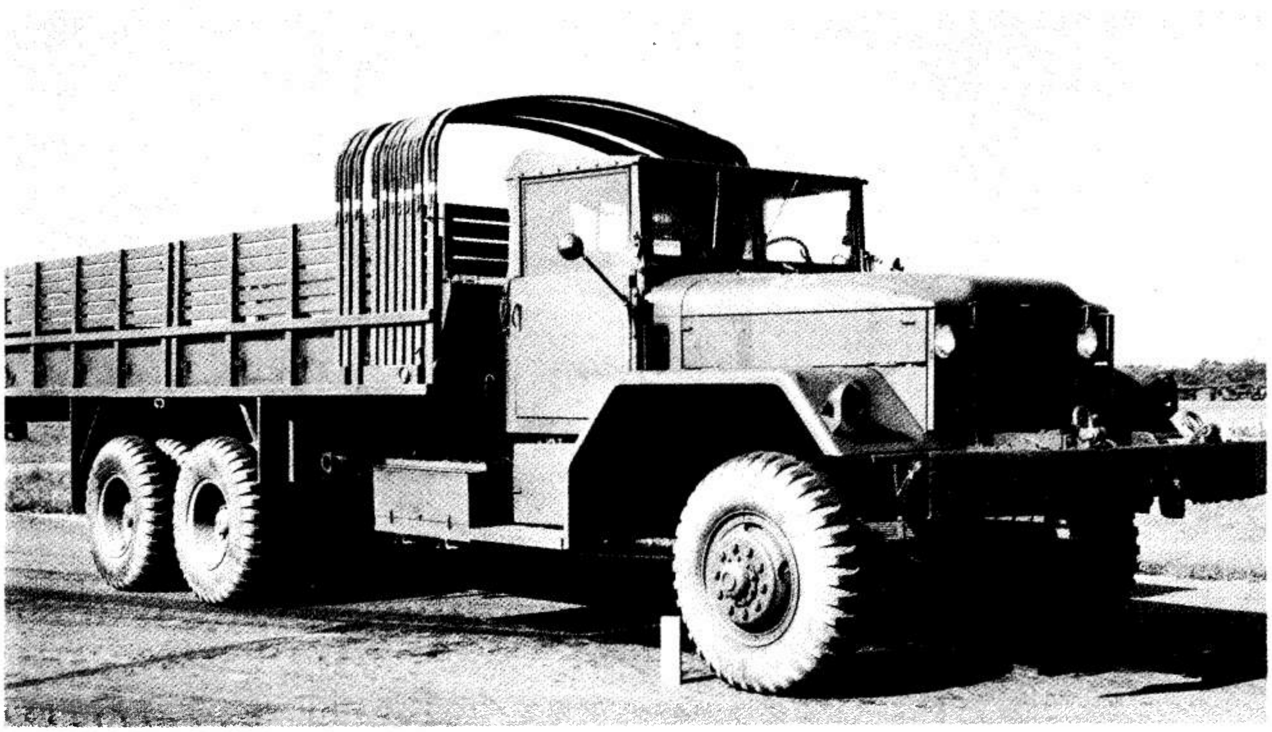


Figure 1-6. Cargo Truck, 5-ton, 6 X 6, M55

Figure 1-7. Cargo Truck, 5-ton, 8 X 8, M656

Figure 1-8. Cargo Truck, High Mobility, 8-ton, 4 X 4, XM520 (Goer)

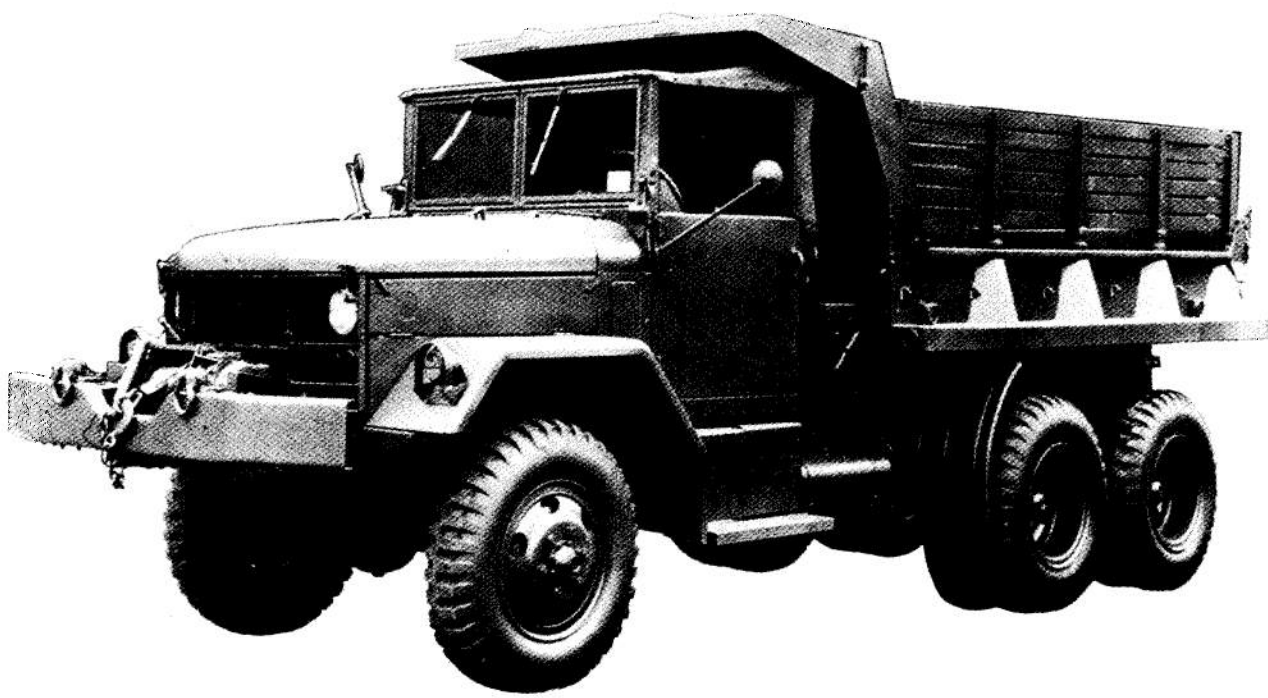


Figure 1-9. Cargo Dump Truck, 2-1/2-ton, 6 X6, M342

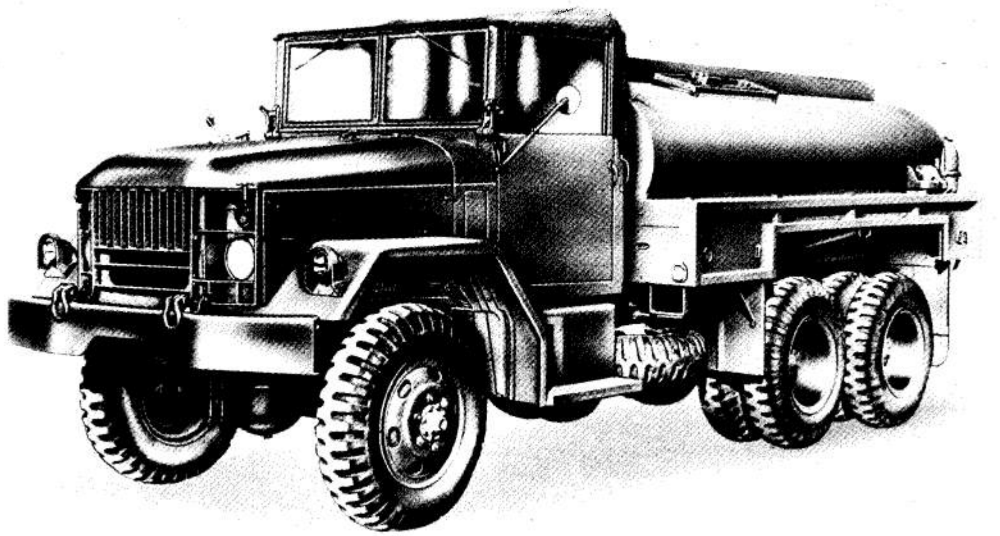


Figure 1-10. Gasoline Tank Truck, 2-1/2-ton, 6 X 6

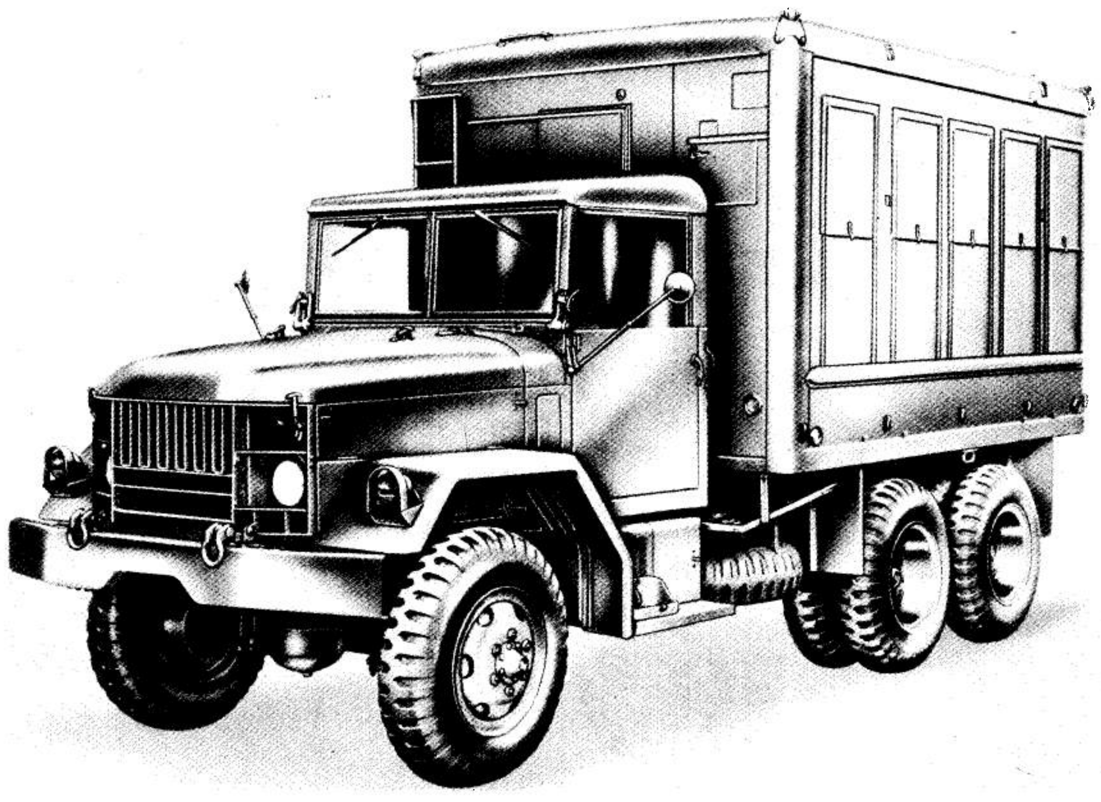


Figure 1-11. Shop Van Truck, 2-1/2-ton, 6 X 6

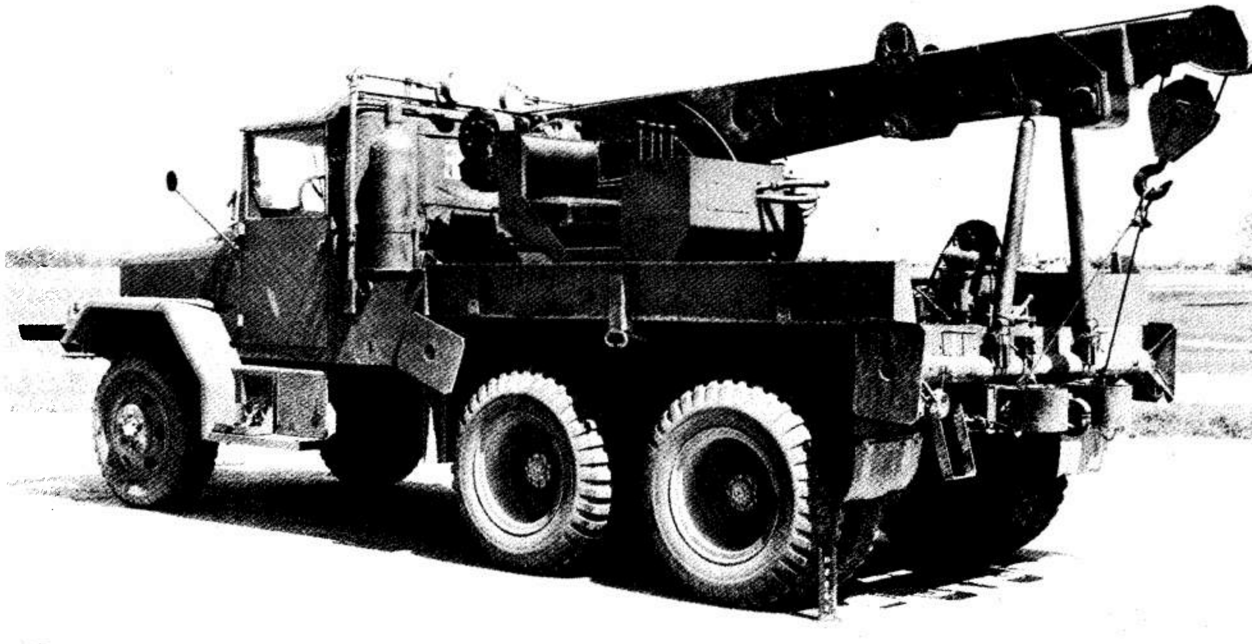


Figure 1-12. Wrecker Truck, 2-1/2-ton, 6 X6, M60

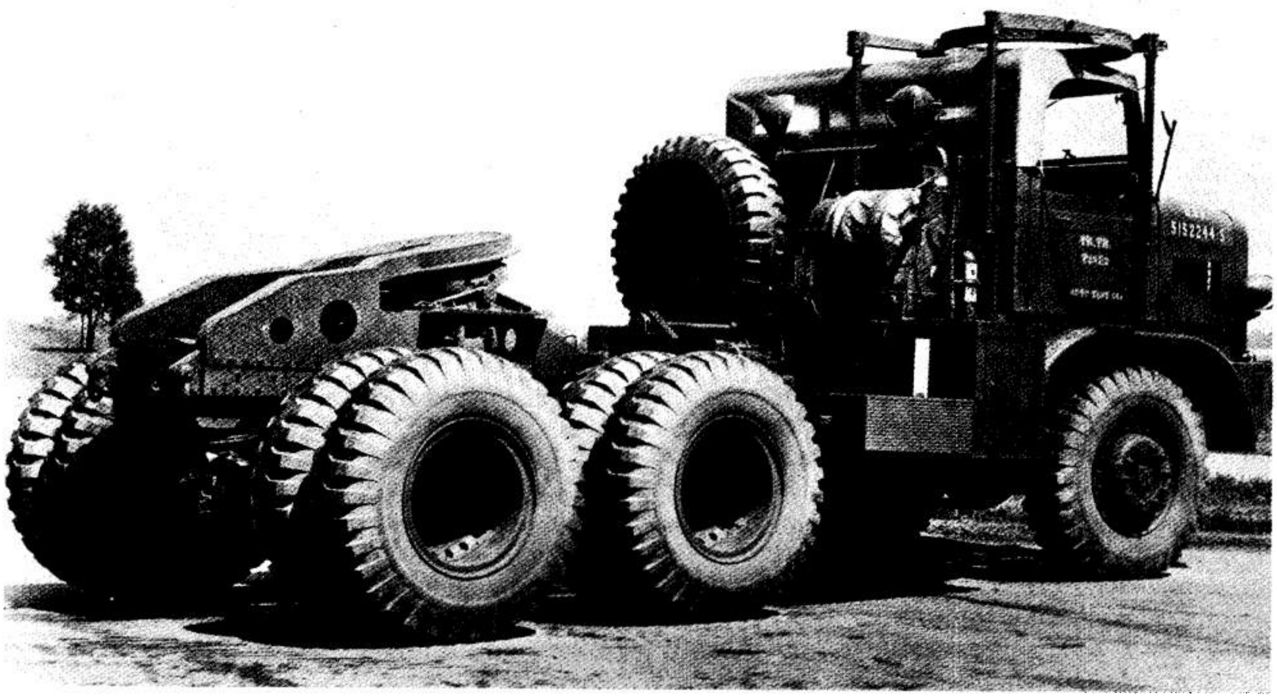


Figure 1-13. Truck Tractor, 8-ton, 6 X6, T28E2

Figure 1-14. Truck Tractor, 12-ton, 6 X 6, M26A1 With Tank Transporter Semitrailer, 50-ton, 8-wheel, M15A2

cab. They are provided with a fifth wheel-type coupler as well as with a towing pintle which makes them capable of towing semitrailers as well as full trailers. Their primary mission is to salvage crashed aircraft and to perform general towing.

Trailers, semitrailers, and tank transporters are unpowered wheeled vehicles designed to be towed by self-propelled vehicles. They are made in the same variety of functional body types as are self-propelled vehicles, and the design considerations and design procedures that pertain to the bodies of self-propelled vehicles also apply to their unpowered counterparts.

The distinction between trailers and semitrailers is that in the former (trailer) all or most of the weight is supported by its own integral wheels; while in the latter (semitrailer) the weight is supported, when in motion, by integral wheels at the rear of the vehicle and by the towing vehicle, or a dolly, at the front through a fifth wheel coupler mounted on the towing vehicle, or on the dolly. When uncoupled from the towing vehicle, or dolly, the front ends of semitrailers are supported by retractable landing gears. A tank transporter (Fig. 1-14) is a heavy duty semitrailer designed primarily to transport tracked combat vehicles. The designer who is concerned primarily with trailers and semitrailers will find an excellent source of information in Ref. 12.

1-4 CHASSIS-BODY RELATIONSHIPS

1-4.1 SEPARATE FRAME AND BODY

The separate frame and body type of vehicle construction (Fig. 1-15) is the most common construction technique employed by vehicle manufacturers, particularly in the field of cargo vehicles. In this type of construction, the chassis frame and the vehicle body are separate entities, each a complete unit by itself. The chassis frame is designed to support the weight of the body and absorb all of the loads imposed by the terrain, suspension system, power plant, drive train, and steering system; while the body merely contains and, in some cases, protects the cargo. The body is generally bolted to the frame at a few discrete points to allow for flexure of the frame and to distribute the loads to the intended load carrying members.

With this type of construction, the body structure need only be sufficiently strong and rigid to contain the weight of the cargo and resist any dynamic loads associated with cargo handling and cargo movement during vehicle operation and to absorb shocks and vibrations transferred from the frame. In some cases, particularly under very severe operating conditions, the body structure may be subjected to some torsional loads that are not completely absorbed by the frame; however, this is not common. This explanation applies,

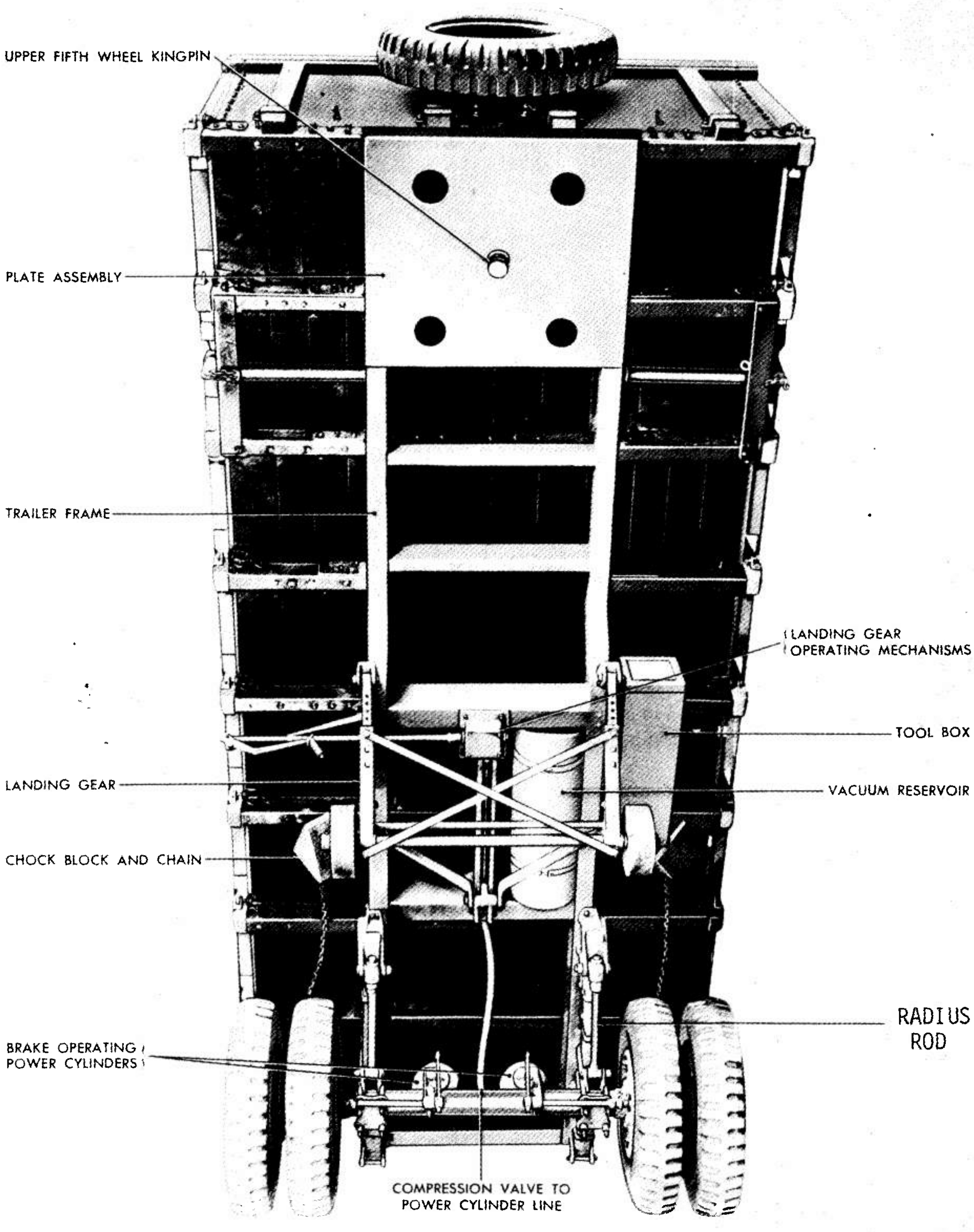
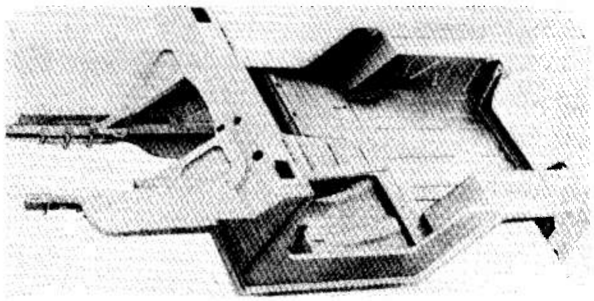


Figure 1-15. Separate Frame and Body Construction



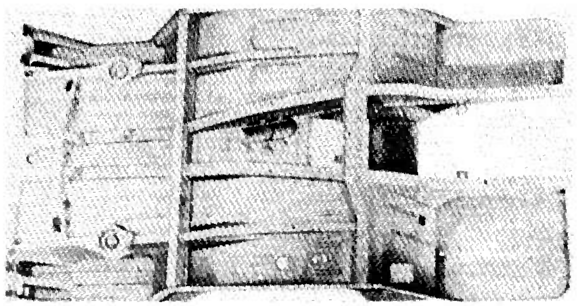


Figure 1-16. Integral Frame and Body, 1/4-ton, 4 X 4, Utility Truck, M151

basically, to heavy trucks and not to passenger automobiles. In a typical passenger automobile, the frame supplies about 37 percent of the torsional rigidity and about 34 percent of the bending rigidity; the balance is supplied by the body structure.

Advantages of the separate frame and body type of construction are:

a. Ease of mounting and dismounting of body structure.

b. Versatility—various body types can be readily adapted to standard chassis.

c. A strong, rugged design is relatively easy to achieve, though at a penalty in vehicle weight.

d. Ground and drive train noise is easily isolated from crew and passenger compartments through the use of rubber mounts between frame and body structures.

e. Relatively simple structure to design and to fabricate, and with simplicity comes economy of manufacture.

f. Racking and twisting of chassis frame are readily isolated from body structure through the use of spring pins, toggle levers, or other semiflexible means of attaching the body to the frame.

Not all is advantageous, however, in the separate frame and body type of construction. The following are some of the more serious disadvantages:

a. Cargo floor is generally higher with this type of construction for the same ground clearance and wheel size.

b. Vehicle silhouette is generally higher when all other factors affecting overall height are kept constant.

c. Vehicle weight is increased.

d. This type of construction is inherently less desirable for amphibious vehicles, because it is less effective in developing maximum buoyancy for minimum weight.

1-4.2 INTEGRAL FRAME AND BODY

The integral frame and body type of construction, also referred to as unitized construction, combines the frame and body into a single, one-piece structure by welding components together, by forming or casting the entire structure as one piece, or by a combination of these techniques. Merely welding a conventional body to a conventional chassis frame, however, does not constitute an integral frame and body construction. In a truly integrated structure, the entire frame-body unit is treated as a load-carrying member that reacts to all of the loads experienced by the vehicle--road loads as well as cargo loads. This is in contrast with the separate frame and body type of construction (par. 1-4.1) in which the frame is the major load bearing element and the body contributes only secondarily to the total strength and rigidity of the combination.

Fig. 1-16 shows an integral frame and body for a 1/4-ton utility truck—the complete truck is shown in Fig. 1-5. Integral type bodies for wheeled vehicles are fabricated by welding together preformed metal panels. The panels are preformed into various load bearing shapes that are so located and oriented as to result in a uniformly stressed structure. Some portions of the integrated structure resemble frame-like parts while others resemble body-like panels. This should not be surprising, since the structure must perform the functions of both of these elements.

Other examples of vehicles with integrated, or unitized, bodies are shown in Figs. 1-7 and 1-8. In these vehicles, the integral construction has been extended to include watertight joints,

sealed doors and access openings, fully enclosed drive train and power plant, watertight cargo and crew compartments, and built-in flotation chambers. The resulting body structure has sufficient buoyance to enable the fully loaded vehicle to float without the need for auxiliary flotation kits.

The concept of a 100 percent uniformly stressed structure is somewhat hypothetical as it is extremely difficult to achieve in practice. Primary loads are generally carried by the underbody rails and crossmembers; while the upper sheet metal, side, end, and roof panels mainly function to enclose and retain the cargo. As the percentage of primary loads carried by the upper parts of the body is increased, the strength and weight of the frame-like components can be decreased correspondingly. This concept has an ideal limit in the monocoque, or single shell, body which is best exemplified by the outer shell of an egg. Automotive bodies have not attained this state of perfection in their body designs as yet. Some racing car bodies, however, and certain single compartment tank-type bodies that have a minimum number of stiffeners do approach perfect monocoque construction fairly closely.

Between the extremes of pure monocoque construction and the simply integrated frame and body is the semi-monocoque body, which is the type that is more practical to achieve. In this type of construction, the sheet metal panels are subjected to the bending and torsional loads experienced by the structure but they are reinforced by stiffening members selectively placed on axes of inherent panel weakness.

For example: a panel member would have considerable inherent resistance to bending moments acting in the plane of the panel were it not for its weakness to buckling (elastic instability) in a plane normal to this plane. The addition of vertical stiffeners, however, will increase the panel's resistance to buckling without reducing its resistance to bending. Stiffeners can be structural shapes (angles, channels, I-sections, etc.), corrugations in the panel, or transverse supports to other structural members. It rests upon the ingenuity of the designer to develop a structure of adequate strength and rigidity at a minimum total weight. Tank and van-type bodies are particularly well suited to the semi-monocoque construction.

Some advantages and disadvantages of the integral frame and body type of construction, when compared to the separate frame and body concept, are:

a. A substantial weight saving is possible in a well designed unitized body.

b. A lower cargo floor and vehicle height are generally possible for equal ground clearance; or greater ground clearance is possible for a given vehicle height.

c. Unitized construction is more suitable for amphibious vehicles and is especially appropriate when maximum protection from mud and water is required for engine, transmission, and drive-line components.

d. Unitized construction is more suitable for a uniform-stress design than is a separate frame and body concept.

e. Unitized construction reduces the amount of vibration present in the vehicle structure.

f. A rigorous analysis may be required to achieve a high degree of weight reduction for unitized structures. An improperly designed unitized vehicle may weigh more than a similar vehicle with a separate frame and body.

g. A greater amount of ground and drive train noise is transmitted into the crew and passenger compartments of vehicles with unitized bodies. The integral construction makes it impossible to interpose sound isolators between body and frame.

h. The mounting of different body types upon a basic chassis is not easily accomplished.

i. To take full advantage of the low floor level possible with the unitized body, it is usually necessary to extend the wheel housings into the cargo area.

1-4.3 HULL-TYPE BODIES

Par. 1-1.3 discusses the various connotations given to the term *hull* in the military automotive field. In summary, the bodies of tracked vehicles are generally referred to as hulls; and the term is also used when referring to the bodies of amphibious wheeled vehicles (Figs. 1-7, 1-17, 1-18), armored cars (Fig. 1-2), and exoskeletal types of vehicles (Fig. 1-8). This last type, incidentally, is also amphibious.

Regardless of whether the enclosing structure of a vehicle is called a body or a hull, its relationship to the chassis is one of the two general relationships discussed in pars. 1-4.1 and 1-4.2; i.e., either a separate frame and body or

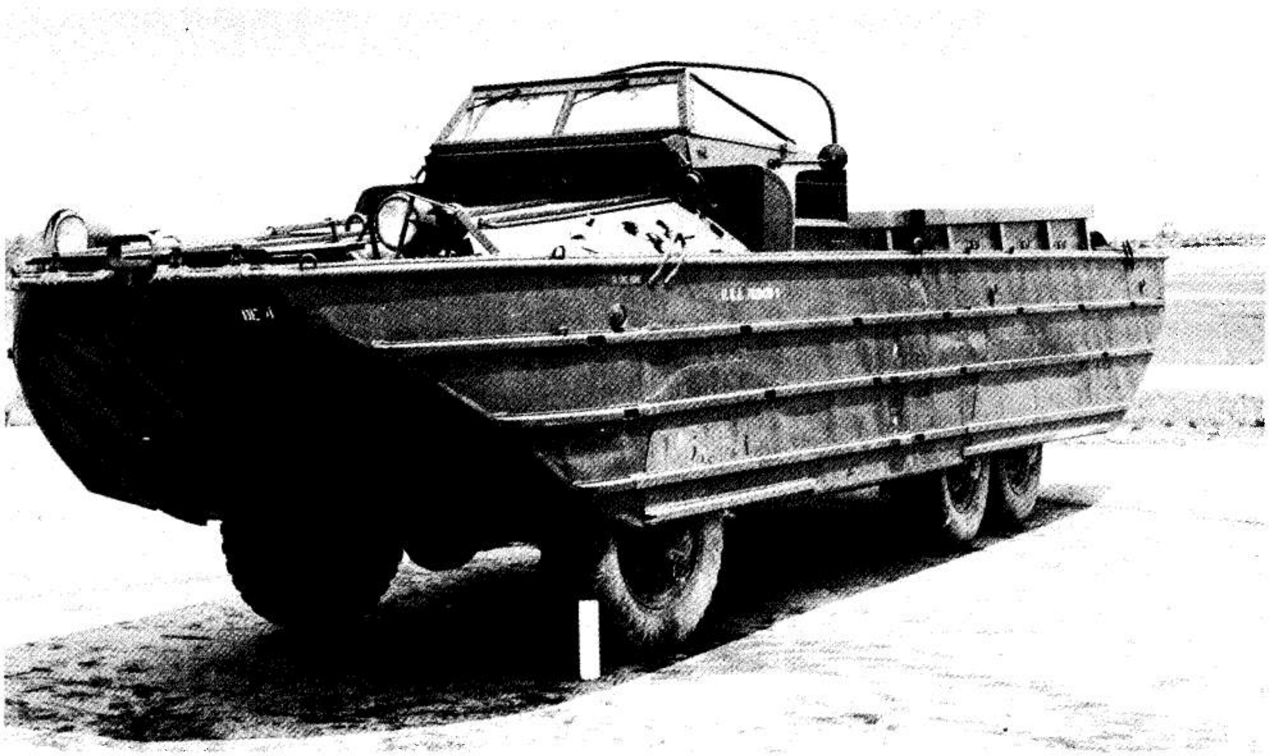


Figure 1-17. Amphibious Cargo Truck (DUKW), 2-1/2-ton, 6 X 6—1954 (APG B593)

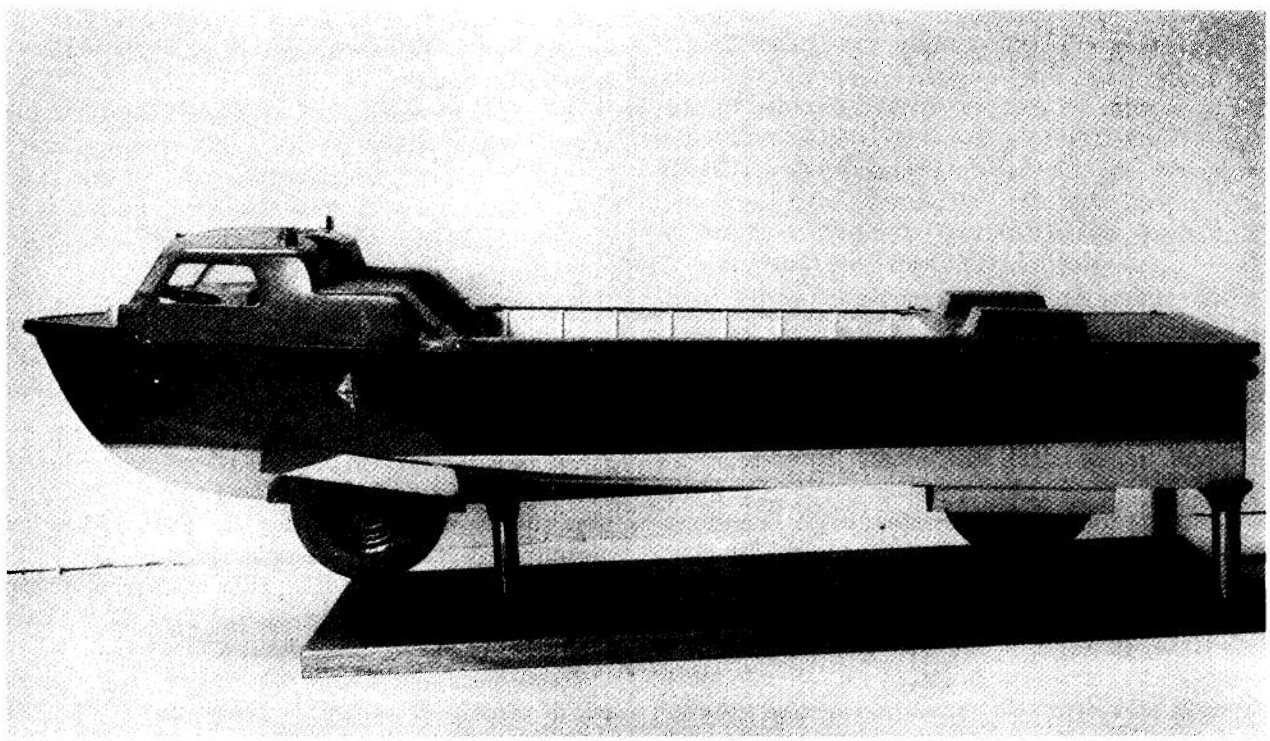


Figure 1-18. High-speed Amphibious Cargo Truck, 5-ton, 4 X 4 (Wheels Lowered for Land Operations)

an integral frame and body. The amphibious truck shown in Fig. 1-17 is designed with a boat-like hull, which is constructed on the separate frame and body principle. Its basic hull is designed to enclose the cargo and crew and provide sufficient buoyancy for flotation and sufficient rigidity while afloat. A frame, similar to a conventional truck frame, is installed inside the hull body and is bolted to it. The power plant and drive train are supported by the frame, and the suspension system is attached to both the frame and the hull.

A contrasting example is the hull of the M60 Tank shown in Fig. 1-1. This is an extreme example of an integral frame and body construction. The heavy armor steel of which it is constructed to resist ballistic impacts and the blast of high explosives imparts so much strength and rigidity to the structure that no additional frame, or frame-like members, is necessary.

1-5 BODY TYPES

1-5.1 MILITARY CARGO BODIES

The military cargo body (Figs. 1-6, 1-7, 1-8) is probably the most common of the military body types. It consists of a shallow box-like structure with an integral front panel, a hinged tailgate, and an open top. In traditional designs such as is illustrated in Fig. 1-6, the sides are integral with the floor and front end, and a hinged tailgate offers access to the cargo space. However, with increasing demands by the vehicle users for easier and more rapid loading and unloading capabilities, designs that incorporate hinged drop-sides on one or both sides of the cargo body, in addition to the hinged tailgate, are under development and are beginning to make their appearance on new vehicle types. In some designs, the hinged side panels extend almost the full length of the body and permit the use of forklift trucks to load and unload the vehicle.

The material most commonly used for military cargo bodies is steel, although aluminum alloys are gaining popularity due to their lighter weight. Floors, or load platforms, are usually flat; but in some designs, in order to achieve a lower floor and vehicle height, wheel housings are allowed to protrude above the floor.

Various accessories attach to the basic body structure. These include removable roof bows which support roll-up tarpaulins and end curtains for cargo and personnel protection, side and end extensions (racks), and removable troop seats. In the traditional military cargo body (Fig. 1-6), the roof bows and side racks are made of wood and are inserted into sockets provided for this purpose in the side and end panels of the cargo box. The troop seats consist of the bottom four or five slats of the side racks. These are hinged to the vertical members so as to form bench-type seats along each side of the body. When not needed for seating personnel, the seats are folded up into the side racks. Cargo bodies with drop sides, currently under development, have similarly hinged troop seats attached to the sides of the body.

1-5.2 VAN BODIES

The van body (Fig. 1-11) is a rigid, fully enclosed, box-like structure with a flat floor and an access door, or doors, in any of its ends or sides. Aside from its obvious capability as a cargo transporter, the van enclosure is especially well-suited to protect its cargo; to provide conditioned air or refrigerated space; to mount and operate electronic instruments; to house medical facilities; and to provide space for shops, office or field command post, or numerous other uses where a completely enclosed space is required.

Traditionally, van bodies were generally frame mounted (par. 1-4). This permitted them to be rather simple structures of sheet metal panels that were usually reinforced with separate framing and stiffening members. Such techniques, however, imposed weight penalties which became undesirable as greater emphasis was placed upon air mobility and amphibious capabilities. This led to developmental work in lightweight van construction—integral frame and body design (par. 1-4.2) combined with stressed-skin structures (par. 4-2.3.3). This became particularly true in the trailer field where van bodies are especially popular. Furthermore, due to the fact that most military van-type trailers are required to move much less frequently than typical self-propelled vehicles (some are even used as semipermanent installations) and are subjected to less rigorous requirements due to a greater flexibility in route selection and speed requirements, it is often

possible to trade-off some durability for reduced weight and cost. This is reflected in the extensive use of aluminum, the appearance of nonmetallic materials, and novel construction techniques that make use of honeycomb and sandwich-type materials.

Since vans are used for a variety of functions, there are many different types of vans in existence; however, most vans can be placed into one of four basic categories, namely, cargo, shop and maintenance, office and command post, and electronic instrumentation. These names indicate the general purpose of the vans in each category. Personnel are often required to perform tasks within the van enclosures and this, plus the necessary equipment and furnishings within the van, makes space availability a problem. As a solution to this, the expansible van body was developed[13]. This design incorporates hinged body sections and cranking mechanisms which permit increasing the lateral dimension of the enclosure by almost 100 percent.

All van-type bodies are basically similar, but large variations in accessory features do exist, and construction techniques and materials used vary considerably between the different models. Sealing out moisture and dust is a serious problem. Seals and sealing techniques are discussed in Chapter 3, Section III. Table 1-1 shows some of the characteristics and equipment that are usually associated with the various van types.

1-5.3 DUMP BODIES

The dump body is an open-top box structure with a flat floor, integral front and side panels, and a hinged tailgate. The body is hinge-connected to the chassis, usually at the rear, and is tilted about this hinge by an elevating mechanism which raises the forward end to discharge loose cargo, as shown in Fig. 1-19. This body type is generally similar to the cargo body discussed in par. 1-5.1, except that the all metal construction is considerably heavier and is characterized by the presence of its own underframe and reinforcing members to stiffen and strengthen the box. The heavier body construction is necessary for the following reasons:

a. High density cargoes are more common to this type of body.

b. Impact loading is common.

c. Body subjected to greater amount of abrasion.

d. Greater possibility of overloading due to heaping loads or concentrating extremely high density cargoes.

e. The cargo floor must be inherently strong and rigid to withstand the dynamic forces to which it is subjected when dumping and spreading its cargo.

Some dump bodies are equipped with a cab protector that is fastened to the front end of the body and cantilevers over the top of the cab. It is of rigid construction to withstand accidentally spilled loads that would otherwise damage the cab and possibly injure ths occupants. When the body is raised for dumping, the cab protector rises with it. They are usually installed semipermanently to permit their removal in low headroom situations, such as loading on board ships or aircraft or clearing railroad overpasses or tunnels.

The tailgate, or endgate as it is sometimes called, is usually hinged at its top rather than at its bottom edge, and the distance to which it can swing open is limited by an adjustable length of chain (see Fig. 1-20). This arrangement limits the rate at which the load can be discharged and is useful when the load is to be spread rather than dumped in one place. The bottom of the gate is released by the operator by means of a lever-actuated release mechanism (Fig. 1-19).

Fig. 1-20 shows a double-acting tailgate. This design offers a choice of under-the-gate discharge, to facilitate spreading a load, or the normal bottom hinge action found on the tailgates of nondumping cargo bodies to permit loading and unloading of cargo over the lowered tailgate. This flexibility is made possible by securing the upper and lower edges of the tailgate to the dump body by means of latch-type hinges. The upper edge of the tailgate is usually released by removing lock pins from the tailgate upper latches which permits the tailgate to swing downward about the lower latches.

1-5.4 TANK BODIES

Tank bodies, as the name implies, are tank-type enclosures that are provided with a means of filling and emptying and are designed

TABLE 1-1 CHARACTERISTICS AND EQUIPMENT COMMONLY ASSOCIATED WITH THE FOUR VAN TYPES

Characteristics and Equipment	*Van Types*			
	Cargo	*Shop-Maintenance*	*Office-Command Post*	*Electronic-Instrumentation*
Windows	No	Some Types	Yes	Some Types
Doors	Full Width, Extra-Large, or Several	Personnel	Personnel	Personnel
Expansible	No	Some Types	Some Types	Some Types
Interior Lighting	Yes	Yes (1)	Yes (1)	Yes (1)
Heating System	No	Yes	Yes	Yes (2)
Ventilating System	No	Some Types	Yes	Yes (2)
Air Conditioning	No	Some Types	Yes	Yes (3)
Refrigeration	Some Types	No	No	No
Insulation	Only With Refrigeration	Some Types	Some Types	Some Types
Blackout Provisions	No	No	Yes	Some Types
Special Elect. Requirements	No	Yes	Yes	Yes

(1) Extensive lighting required and usually both 24 volt DC and 120 volt AC systems required in same vehicle.
(2) Special temperature control often required.
(3) Heavy air-conditioning load common due to electronic equipment.

to be readily attached to vehicle chassis. Their most common application in the Armed Forces is for transporting liquid cargoes such as water, gasoline, other liquid fuels and petroleum products; and compressed gases. They are also used to transport corrosive fluids, chemicals, foodstuffs, and a variety of dry materials; although these are not common, at present, as military applications. The shapes of these tank enclosures vary widely as designers endeavor to attain a low center of gravity, low overall vehicle height, minimum vehicle length, minimum shifting of cargo, and adequate rigidity with minimum weight. Tanks for dry, powdery cargoes have the additional problem of preventing packing which hampers rapid unloading. Fig. 1-21 shows some typical tank body shapes. A typical tank body for transporting gasoline is shown in Fig. 1-10.

Although the appearance of tank bodies may vary considerably, structurally they are quite similar. Physical and chemical properties of the cargoes for which specific bodies are designed influence such considerations as the materials that are used, wall thicknesses, reinforcing members, compartmenting bulkheads, types and sizes of access openings, loading and unloading facilities, cleaning provisions, and safety

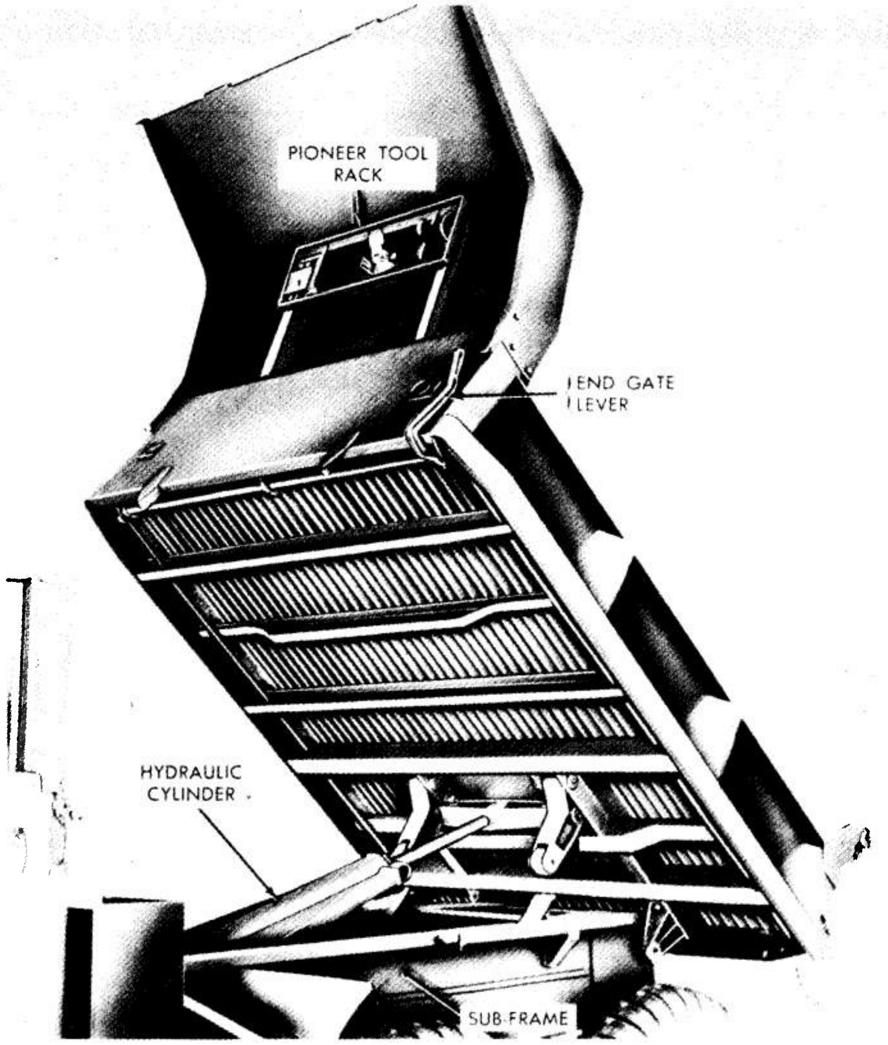


Figure 1-19. Typical Dump Body in Raised Position

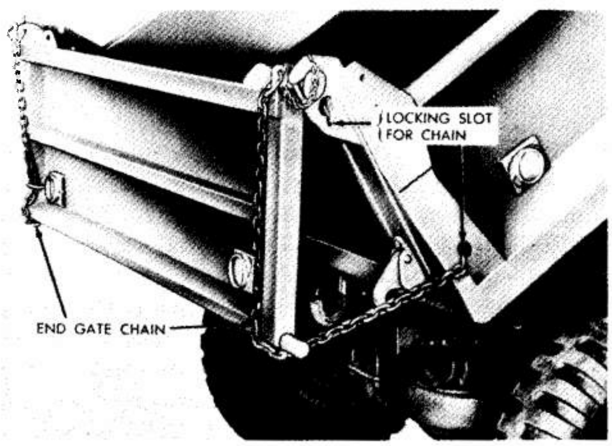


Figure 1-20. Dump Truck Tailgate, Opened for Dumping

equipment required. Since many tank transported commodities are potentially hazardous, their movement on public highways is subjected to safety regulations. These are discussed in par. 4-17.1.

Some tank bodies consist of a single compartment while others are divided into a number of separate compartments by suitable bulkheads. The bulkheads add strength and rigidity to the structure and serve to limit surging of liquid cargoes due to motion of the vehicle; however, they increase the piping requirements and the dispensing equipment necessary. A series of baffles is also used to limit surging or sloshing of liquid cargoes. These baffles are so devised as to leave the various chambers interconnected, thus eliminating the

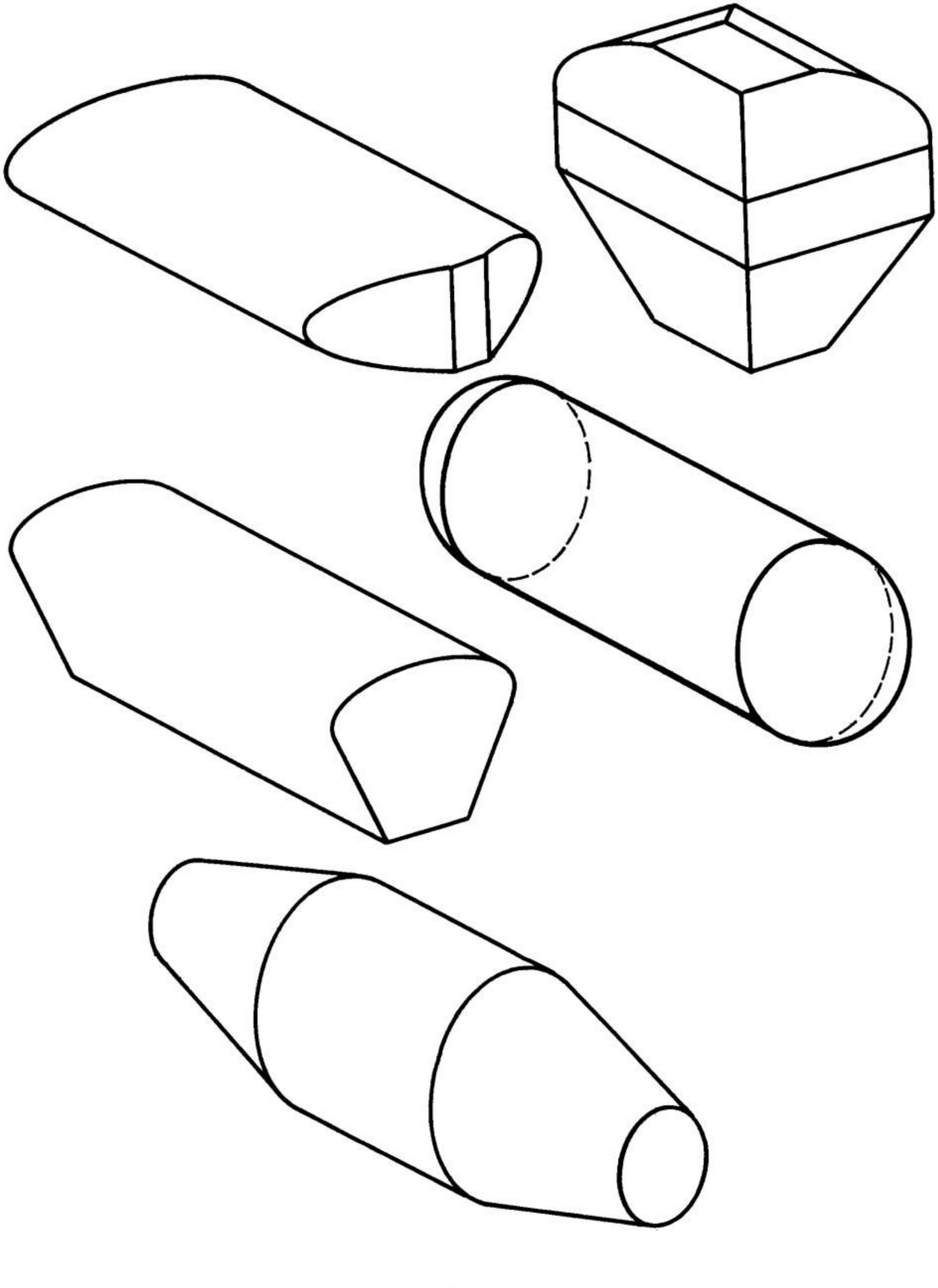


Figure 1-21. Some Typical Tank Body Shapes

need for filling and unloading separate compartments individually.

Steel and aluminum are primary construction materials for tank bodies; but Fiberglas-reinforced plastics have been used quite successfully for water tanks and tanks carrying nonhazardous cargo. Since keeping water at reasonable temperatures for drinking (above freezing and below 80°F) is a major problem in mobile tanks[14], the insulating properties of the plastic materials are quite attractive. This is especially true for tank trailers which are often left unattended in remote locations. In cold areas, heaters are used to keep piping, pumps, and valves from freezing.

Tank bodies for flammable liquids present less insulation problems than do water tanks because of their much wider temperature usability range. However, they present other requirements, such as fire extinguishing equipment, static electricity dischargers, and venting provisions.

Acid transporting tanks require special considerations in several areas such as material compatibility, protective coatings, special seals, overturn protection around access openings, and heating requirements for caustic solutions to prevent their crystallization during transit.

Tanks for transporting liquid gases are subjected to additional requirements. Among these is the need for thermal insulation to minimize evaporation and a structure designed to withstand internal pressure. In general, this involves greater wall thicknesses, pressure tight joints, and fewer and smaller openings for access and discharge.

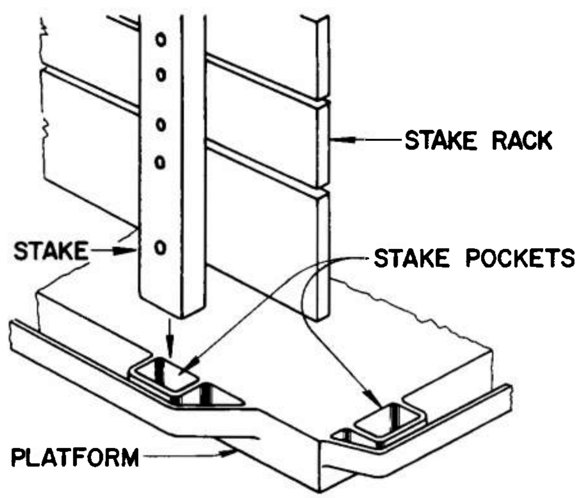
Detailed information on the requirements of many tank-transported products is given in Ref. 15.

1-5.5 PLATFORM BODIES

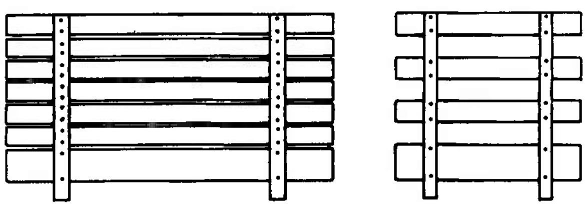
The platform body is the simplest of all the body types—it is so simple that it is questionable whether it should be even called a body. But, since the body of a vehicle is defined as that assembly of components that constitutes the cargo- or equipment-carrying portion of a vehicle in accordance with its functional purpose (par. 1-1.3.3), the platform satisfies this definition and is, therefore, a type of body. It consists of a flat load-bearing deck or floor that is attached to, and is supported by, the vehicle chassis. An illustration of this type of body is shown in Figs. 1-4 and 1-14. Flat-bed trailers, low-bed semitrailers, and certain types of wrecker trucks are equipped with this type of body. The platform deck is usually covered with hard wood planking to provide a resilient and readily replaceable wear surface.

1-5.6 STAKE AND PLATFORM BODIES

This body type, as the name implies, is very similar to the simple platform body described in the preceding paragraph. It is a platform body that is constructed with stake pockets around its perimeter into which stake racks are inserted to complete the body (Fig. 1-22(A)). Stake racks consist of a series of horizontal slats bolted at right angles to vertical members called stakes, as illustrated in Fig. 1-22(B). The racks are made in



(A) STAKE AND PLATFORM INTERFACE



(B) TYPICAL STAKE RACKS

Figure 1-22. Details of Stake and Platform Body

various lengths and are connected together to form cargo retaining sides and ends atop the platform. Various methods are used to fasten the sections to each other—chains, hinges, latches, hooks, etc.

Operationally, stake and platform bodies are similar to cargo bodies. In loading or unloading certain types of cargo—such as heavy machinery, large slabs of stone, steel plate, large rolled steel sections, large beams, and other very bulky cargo—the permanent sides of the typical cargo body are a hindrance that is eliminated with the platform body. The addition of the stake racks adds some versatility to the simple platform body.

1-5.7 BODY-ON-CHASSIS BODIES

The body-on-chassis type of body construction is rather uncommon on military vehicles but is perhaps the most common on civilian vehicles, since it includes sedan-type passenger car bodies. Par. 1-1.3.3, in defining vehicle bodies, states that the body excluded the chassis assembly and, with the exception of a few special cases, also excluded the vehicle cab. The few special cases referred to were those vehicles which have a body-on-chassis construction. This includes such vehicles as sedan cars, busses, some ambulances, and the 1/4-ton utility vehicles. The main difference between these vehicles and the conventional military truck is that they do not have a distinctly separate cab. The body-on-chassis type of body comprises the entire structure above the chassis frame including the hood, instrument panel, and driver's compartment. For illustrations of this concept, compare the integrated hood-cab-body of the vehicles shown in Figs. 1-3 and 1-5 with the separate cab types shown in Figs. 1-6, 1-9, 1-10, and 1-11.

The main attraction of this type of body is that a more efficient utilization of space is possible with an accompanying savings in overall weight. This design, however, is only suitable for vehicles in which it is desirable for the operating personnel to have ready access to the cargo compartment.

SECTION III—TYPES OF TRACKED VEHICLES

1-6 GENERAL DESCRIPTION

Tracked or track-laying vehicles, as a type classification of land vehicles, include all vehicles that move about on flexible tracks rather than on wheels. The principles of this type of suspension system are thoroughly discussed in many texts[6,10]. Suffice it here to state merely that the concept dates back to about 1770. It is based upon the use of a flexible roadway, fashioned in an endless loop, and carried along by the vehicle. This the vehicle places continuously upon the ground, travels over it, and picks it up again.

In some respects, this type of suspension system is similar to that of wheeled vehicles. Like a wheeled vehicle, the tracked vehicle rolls along on a number of wheels, referred to as road wheels, that are arranged in tandem along each side. The mass of the vehicle is elastically supported on this system of road wheels in much the same manner as is the sprung mass of a wheeled vehicle. From this point on, however, several differences exist. Instead of rolling on the ground, as the wheels of wheeled vehicles do, the road wheels roll on the inner surfaces of the track loops. Tractive effort is developed by the interaction of the tracks with the ground, and the propulsive effort is applied to the tracks by means of driving sprockets at one end of the track loops instead of being applied to the wheels as is the case with wheeled vehicles. The main function of the tracks is to distribute the weight of the vehicle more uniformly on the ground and over a larger area. This results in a lower ground pressure and improved mobility, particularly in soft soil. Tracked vehicles are generally considered to have superior terrain crossing capabilities to wheeled vehicles; although, the gap is being narrowed by wheeled vehicle research. A high price is exacted for the improved cross-country capabilities of tracked vehicles in the form of greater power losses to drive the tracks and to steering (tracked vehicles are difficult to steer), a relatively short life for tracks and suspension components, relatively low speed capabilities on improved roads, and a high monetary cost of tracks.

As has been stated previously (par. 1-1.3), the enclosing structures of track-laying vehicles are usually referred to as hulls rather than bodies. Hull types and their construction vary considerably, being influenced by the basic function or mission of each vehicle type concerned; and yet, there is a certain amount of similarity between them. This is largely due to the fact that most track-laying vehicles are either combat vehicles or tactical vehicles that are designed to accompany combat vehicles so closely as to be almost confused with them. Thus, track-laying vehicles are often—though not always—armored, armed, and generally present a rather indestructible appearance. The various types found in the Armed Forces, their missions, and hull design considerations are discussed in the paragraphs which follow.

1-7 HULL TYPES

1-7.1 ARMORED HULLS

1-7.1.1 Combat Tanks

A combat tank, generally referred to simply as a tank, is a self-propelled, armored vehicle designed for offensive combat in either nuclear or non-nuclear warfare (Fig. 1-23). Current tanks are equipped with high-velocity, direct fire guns capable of defeating enemy tanks, bunkers, and providing effective supporting fire for infantry assault. Future tanks may be equipped with launchers for missiles as either their primary or secondary armament or with dual purpose weapons capable of functioning as either tube-type artillery or as launchers for missiles.

The tank hull is the strongest and heaviest hull used on any vehicle. It must be fabricated to withstand ballistic shocks from high-velocity kinetic energy-type projectiles, the blast effects of high explosive rounds and mines, the penetrating effects of chemical energy rounds, and be sufficiently rigid to provide a stable firing platform for the primary weapon system and fire control equipment. In addition, it must provide sufficient space in its interior to house the propulsion system, an adequate fuel supply, working space for the crew, and stowage space for ammunition and necessary supplies and equipment. To top-off these requirements, the hull must be sufficiently small and light to have a reasonable degree of maneuverability and to be able to pass over and through standard traffic

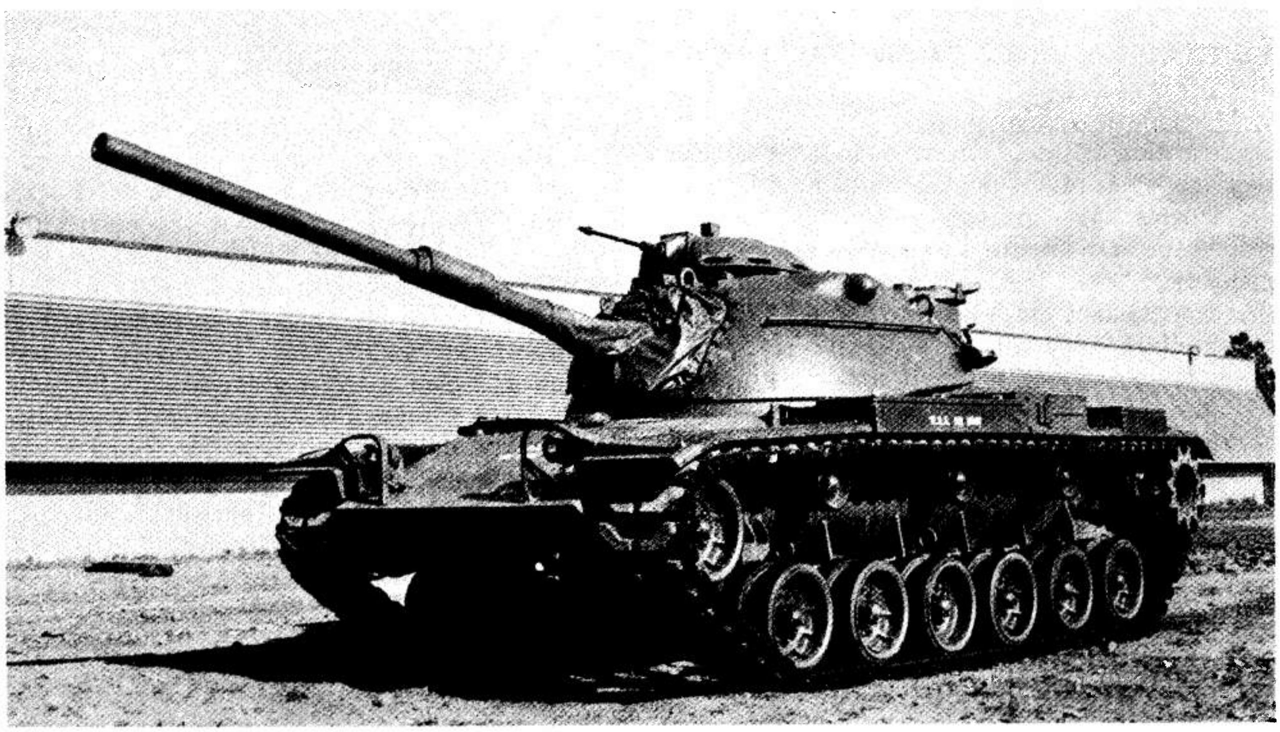


Figure 1-23. Combat Tank, M60, 105 mm Gun

lanes, bridges, and overpasses. These requirements are discussed in greater detail in Chapter 2. The typical tank shown in Fig. 1-23 weighs over 50 tons and is more than 10 ft high by 23 ft long exclusive of the gun, while the empty hull alone (Fig. 1-1) weighs in excess of 14 tons.

Tank hulls of the type shown in Fig. 1-1 are one-piece castings of homogeneous armor steel with a welded floor of rolled armor plate. They contain a driver's compartment in the front portion, a crew compartment at the center, and a compartment for the engine and transmission at the rear. The crew compartment is separated from the engine compartment by a steel bulkhead that is welded in place and which serves as a firewall between the two compartments. Access doors and plates in this bulkhead provide access to the accessory end of the engine.

The engine and transmission compartment at the rear of the hull is provided with a removable top deck which is secured to the hull and to lateral beams. A number of hinged doors and bolted-on plates on the top deck give access to various parts of the engine and transmission. In addition, sections of the top deck can be removed for greater access; and, in the extreme case, the entire top deck can be removed as a unit. Additional access plates are provided in the rear and bottom of the hull for access to service areas of the transmission and engine.

A driver's hatch and door are provided in the top of the forward portion of the hull directly above the location for the driver's seat. An emergency escape hatch and door are provided in the bottom of the hull beneath the driver's seat location.

The top of the center portion of the hull is fashioned into a large ring into which is mounted the turret race ring. The turret, which is not considered a part of the hull, rotates on ball bearings in the turret race ring. The front of the hull slopes and curves backward on the top, bottom, and sides to increase its obliquity to frontal attack.

Casting a tank hull requires a considerable amount of labor and time—labor to accomplish the extensive molding, pouring, and checking of the finished casting and time to allow for slow cooling of the poured metal and for slow stress relieving and annealing. An obvious alternative to the casting of tank hulls is to fabricate them of rolled plates of the proper configuration. Ballistic considerations, however, forbid the use of welded seams in certain critical areas such as the nose, for example. In addition, the desired variations in the wall thickness within the nose section are such that the use of a welded plate construction would present an inordinate number of weld seams, or would result in the use of a compromise of equalized section thickness that would add substantially to the weight of the vehicle. However, investigations have shown[17,18] that the closed die forging process is a feasible method for the production of tank nose sections. The remainder of the hull can be formed of fabricated plate using conventional plate shaping, trimming, and welding techniques. This manufacturing method should offer substantial advantages in terms of manufacturing costs, lead time, improved metallurgical properties, and production rates and may become the method used for the manufacture of future tank hulls.

1-7.1.2 Self-propelled Artillery

Self-propelled artillery comprises a family of weapon systems consisting of an artillery piece permanently mounted upon some form of automotive land vehicle along with all necessary fire control, service, and related equipment necessary to enable the system to operate as a combat unit (Fig. 1-24). The artillery piece can be either the customary tube type or a rocket launcher; and the complete weapon system can be designed expressly for missions of antiarmor (antitank), antiaircraft, conventional support for infantry, armored, or airborne units, or for a combination of these roles. The primary weapon is fired from the vehicle, since it is permanently mounted upon it; and the vehicular mounting gives it the capability of displacing rapidly under its own power. Considerable emphasis in the design of these weapon systems is placed upon mobility; and since weight is a serious handicap to the mobility of vehicles, liberal use is being currently made of aluminum alloys, even for armor protection.

There is considerable similarity between combat tanks and self-propelled artillery both in appearance and in the general nature of the weapon system. Both are basically a weapon system mounted upon a mobile platform. Their main differences, however, stem from

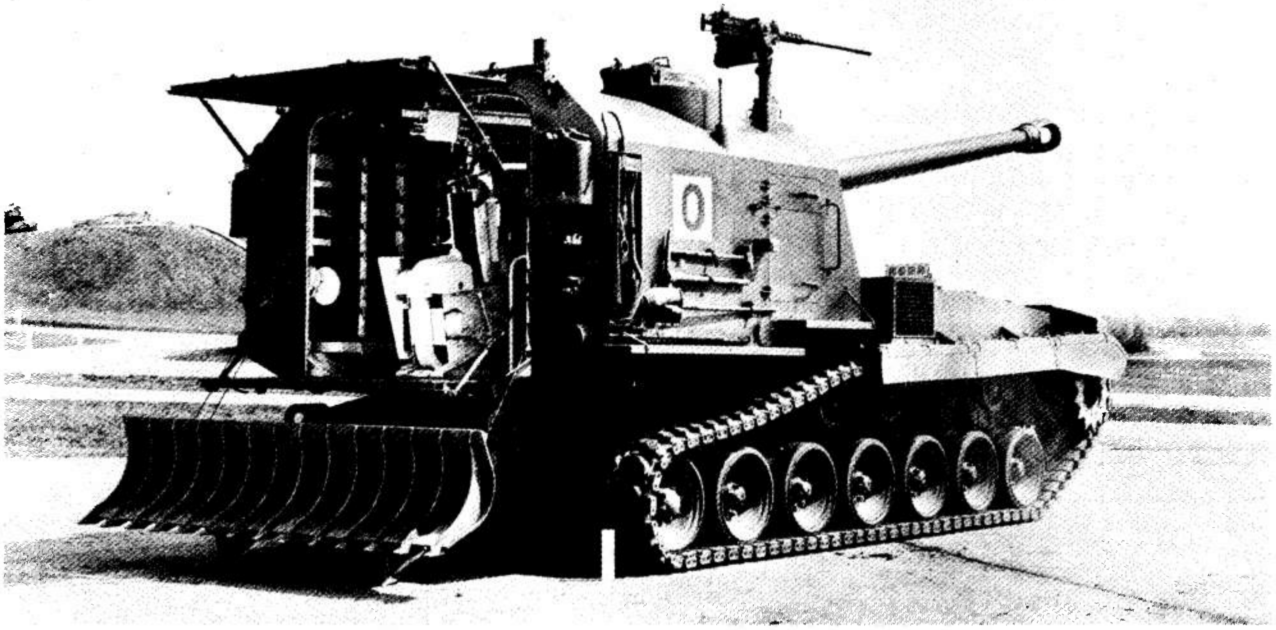


Figure 1-24. Self-propelled Gun, 155 mm, M53

differences in their missions. The tank is primarily a highly mobile assault weapon designed for rapid deployment against the enemy, including his tanks. Its mission is to close with and destroy enemy forces through the use of fire, movement, and shock action; and to this end, it is designed to shoot, to be shot at, and to move as rapidly as possible across all types of terrain. Tanks, therefore, are heavily armored and are provided with a weapon system designed primarily for direct fire missions rather than indirect fire. Both the primary and secondary weapons of the tank are served while the crew is "buttoned up" inside the vehicle. While it is physically possible to load, aim, and fire the primary weapon while the tank is in motion, this is seldom done for reasons of firing accuracy. The usual procedure is to load the weapon and take preliminary aim while the tank is in motion, then stop momentarily to adjust the aim and to fire.

Self-propelled artillery, on the other hand, has the basic mission of all artillery—to support the ground action by heavy concentrations of well directed fire—and brings to this basic mission the advantage of being fully mobile under its own power. The crew usually dismounts and takes up action stations when the weapon is put into action; and, since artillery is not normally used in direct confrontation with the enemy as tanks are, only sufficient armor is required to give protection from small arms and projectile fragments. This easing of space and ballistic protection requirements allows a lighter weight hull to be used, permits a more advantageous location of power plant and transmission, makes possible the installation of a more effective weapon and more sophisticated fire control equipment, and provides additional space for ammunition storage.

Typical characteristics of self-propelled artillery, particularly in the heavier gun types, are: the power plant, transmission, and final drive are all usually located at the front of the vehicle rather than at the rear; the gun is mounted upon an open turret which, in some models, is enclosed by a large cab structure (Fig. 1-24); the principal weapon has a limited traversing capability (about $\pm 30°$); the cab and body are lightly armored in contrast to the heavy armor found on tanks; a hydraulically operated recoil spade, which resembles the blade of a bulldozer, is provided at the vehicle rear to help stabilize the vehicle when firing. There are

other differences, but these are the most conspicuous.

Initially, self-propelled guns were made by adapting standard field artillery pieces to modified tank chassis. This system proved inadequate, however, since the full potential of the system could not be realized due to limitations imposed by the basic characteristics of the two major components. As a consequence, present day self-propelled guns comprise weapons and chassis specifically designed for the mission requirements. While the self-propelled gun vehicle uses many tank components, such as the engine, drive train, and suspension components, the heavy tank hull has been replaced by lighter hulls.

Fig. 1-24 shows one type of self-propelled artillery vehicle mounting a 155 mm gun. The gun and carriage assembly (with the exception of the barrel) is enclosed in a cab fabricated of welded homogeneous steel armor plate. A crew of six is housed inside the cab and hull while traveling. The hull is a welded structure of homogeneous steel armor plate and is divided into four compartments: engine and driver (at front), fuel tank, turret support, and storage. The engine compartment is decked over to provide protection from falling projectiles and the elements. The gun turret is mounted in the turret support compartment and houses the gun mount. The particular self-propelled weapon illustrated has limited on-carriage traversing capabilities (±30°) because of the fixed cab design. In some designs, however, the gun is mounted in a fully enclosed, fully rotatable turret which enables the gun to be traversed through 360°. The total weight of the vehicle shown in Fig. 1-24 is approximately 96,000 lb. It is approximately 22½ ft long, 11¾ ft wide, and 11¾ ft high excluding the gun and recoil spades. With the gun and spade rigged for traveling, this weapon system has an overall length of 33½ ft.

There are other types of self-propelled artillery vehicles both larger and smaller than that shown in Fig. 1-24. Some have fully enclosed, fully rotatable turrets, and some have no turrets at all. A more complete description of the various types is given in Refs. 6, 9, 10, and 11.

1-7.1.3 Armored Personnel Carriers

Armored personnel carriers (Fig. 1-25) constitute a family of vehicles intended primarily to add mobility to the infantry in the zone of combat, particularly to the infantry units of the mechanized and armored divisions. They are high-speed, armored, track-laying vehicles designed to provide protection from small arms fire, overhead artillery bursts, and fragments from explosive projectiles, bombs, and mines. Their interiors are heated and ventilated to provide some degree of comfort for the personnel inside; and large, quick-opening doors at the rear permit rapid entrance and exit. Light armament is sometimes provided on the vehicle to provide defense against aircraft and fire support to the deploying infantry. A pintle or towbar at the rear permit the vehicle to tow a cargo trailer, artillery weapon, or a disabled vehicle when necessary. Current vehicles of this type are given amphibious capabilities through the use of lightweight, watertight hulls, selective load distribution to attain correct trim in the water, and a means of water propulsion and steering. The latter is usually accomplished by using the tracks as paddle wheels and shrouding the returning portions of the tracks to minimize their negative thrust. In this manner, water speeds of 3 to 4 mph are attained in calm, still water. Screw propellers and hydrojets may be used when better water performance is desired.

Thus equipped, the armored carriers can advance well toward the forward edge of the battle area before having to dismount the troops. When not transporting troops, these vehicles are well-suited for carrying ammunition and other cargo. Adaptations of these vehicles are used as mobile command posts and mobile firing platforms for mortars, antiaircraft weapons, and missiles.

The armored personnel carrier shown in Fig. 1-25 has a total weight when loaded for combat of about 24,000 lb, including a driver, troop commander, and 11 fully equipped men. It is fully amphibious (3.7 mph water speed), transportable by aircraft, and can be parachute-dropped. It is provided with an open-type cupola at the troop commander's station, equipped with a pintle-mounted caliber .50 machine gun capable of 360° rotation. The vehicle is approximately 16 ft long, 8.8 ft wide, and 7.2 ft high exclusive of the gun. The hull is a welded structure of aluminum alloy armor plate. Depending upon the degree of ballistic protection desired (a determining factor in the

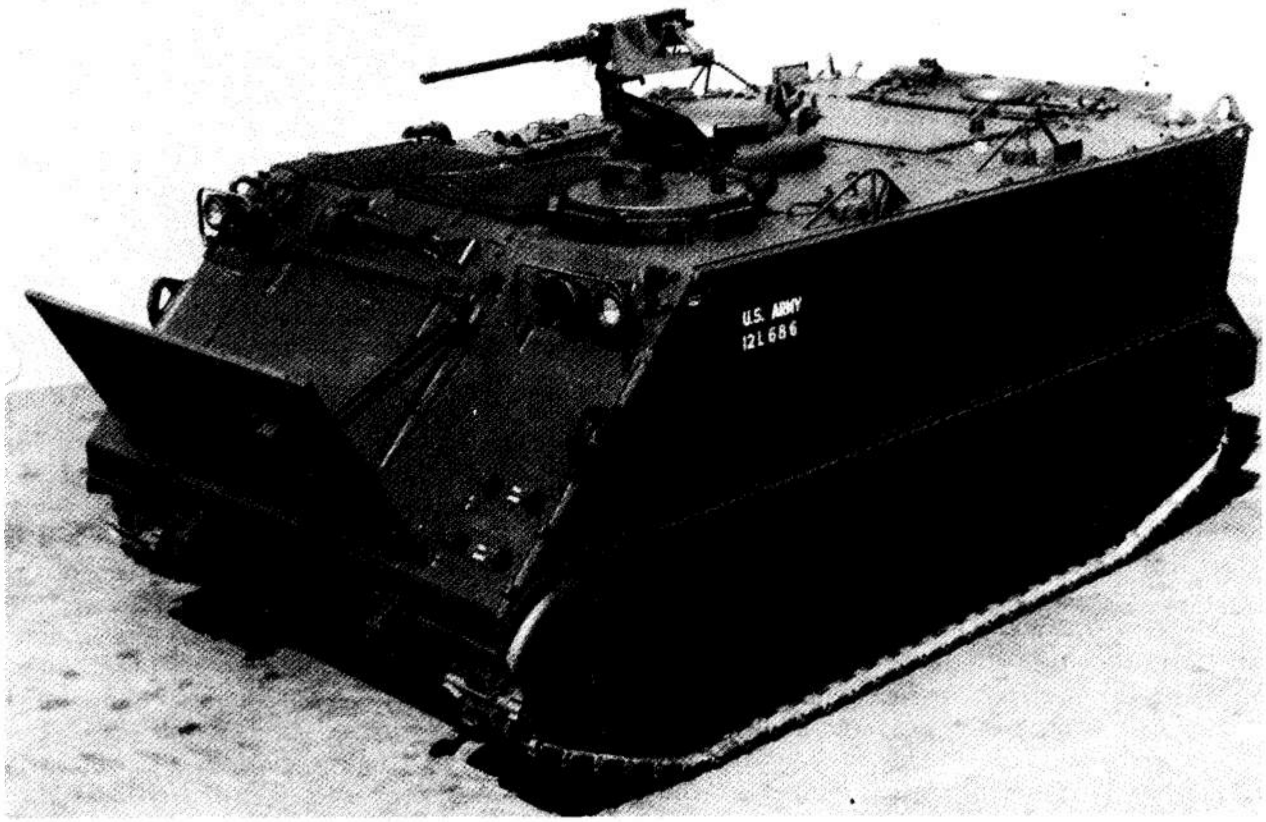


Figure 1-25. Armored Personnel Carrier, M113

plate thickness required), the hull construction can be essentially frameless, as is the case with tank hulls; or it can be a frame and skin structure to minimize the overall hull weight.

1-7.1.4 Armored Recovery Vehicles

Recovery vehicles are fully tracked, self-propelled, armored vehicles designed to rescue disabled tanks, self-propelled artillery, and other mechanized equipment from the battlefield. Toward this end, they are equipped with heavy duty engines, large capacity cranes and winches, and hydraulically operated spades at one or both ends with which to anchor themselves against the pull of their winches, to support themselves when lifting heavy loads, or for clearing sites for recovery operations. These vehicles are also used to lift engines, transmissions, artillery gun barrels or gun assemblies, and other heavy components during the repair of disabled vehicles in the field.

Originally, recovery vehicles were merely tanks with their primary weapons, fire control equipment, and other removable items replaced by cranes, winches, and related equipment. These modified tanks were replaced by vehicles specifically designed for recovery missions; however, in keeping with the Army's general standardization policy, many of their major components—such as power plants, power trains, tracks, and suspension components—are common to other vehicles. A more recent trend is to utilize the chassis designs of self-propelled artillery for conversion to recovery vehicles. Cranes and winches are usually mounted on fixed turrets, especially in the medium- and heavy-duty series of recovery vehicles; although, some light recovery vehicles (M578) do have their cranes and winches mounted in fully rotatable turrets.

The vehicle hulls and cabs are usually constructed of welded steel plate with light steel armor plate enclosing the areas occupied by the

crew and by the engine and transmission. Defensive armament, only, is provided usually in the form of a caliber .50 machine gun for antiaircraft protection. The vehicle can be completely "buttoned-up" and operated in this condition with the crew protected from small arms, mortar fire, and land mines.

Fig. 1-26 shows one type of heavy recovery vehicle. It is designed to recover disabled tanks and mechanized artillery weighing 50 to 60 tons. This vehicle is equipped with a 30-ton capacity crane, a 45-ton capacity main winch, and hydraulically operated spades fore and aft. The vehicle is 33 ft long, 12 ft wide, 10¾ ft high, and weighs about 60 tons.

1-7.1.5 Combat Engineer Vehicles

Combat engineer vehicles have been known by various other names such as pioneer tanks, engineer armored vehicles, and engineer tanks. The current official name for these vehicles, however, is full-tracked combat engineer vehicles. These vehicles are essentially combat tanks that have been modified to accept various engineering attachments. Their purpose is to transport engineer personnel and munitions across fire-swept areas to accomplish essential pioneer missions in support of armored formations. The engineering attachments normally include a heavy A-frame boom (some models have two), a heavy duty winch, and a bulldozer. The A-frame booms have been mounted to the hulls in some models (M102), and to the front of the turret in others (M728). With the boom turret mounted, the winch is installed in the bustle of the turret. All three attachments are hydraulically operated.

The primary armament of current combat engineer vehicles is a 165 mm Demolition Gun (XM135) which replaces the primary tank gun in the turret. It fires an unconventional high explosive plastic projectile to a range of approximately 1,000 yd and is intended primarily for breeching reinforced concrete fortifications.

The principal considerations when modifying a tank hull for conversion to a combat engineer

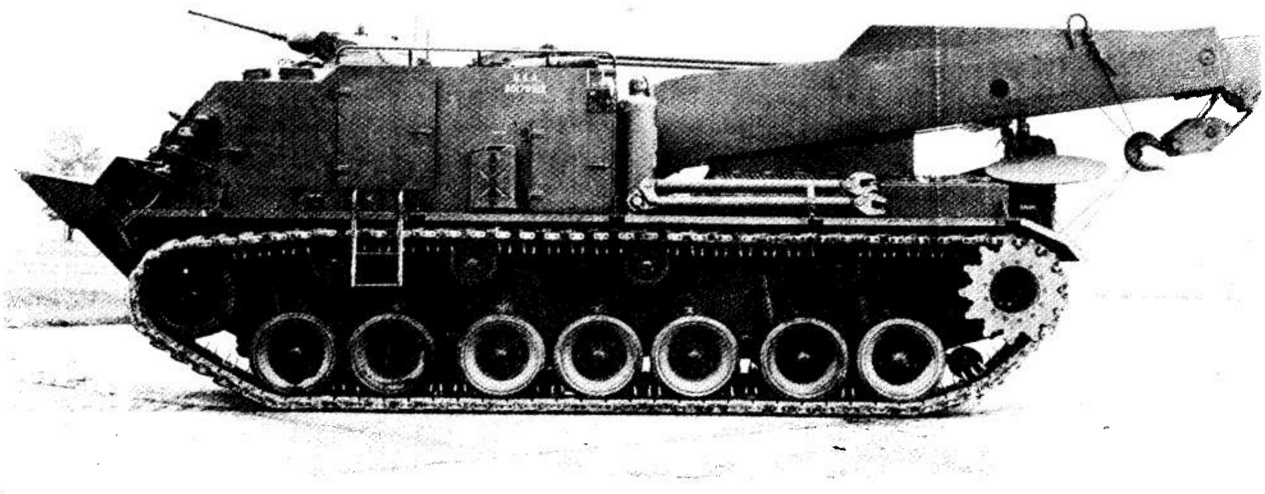


Figure 1-26. Heavy Recovery Vehicle, M51

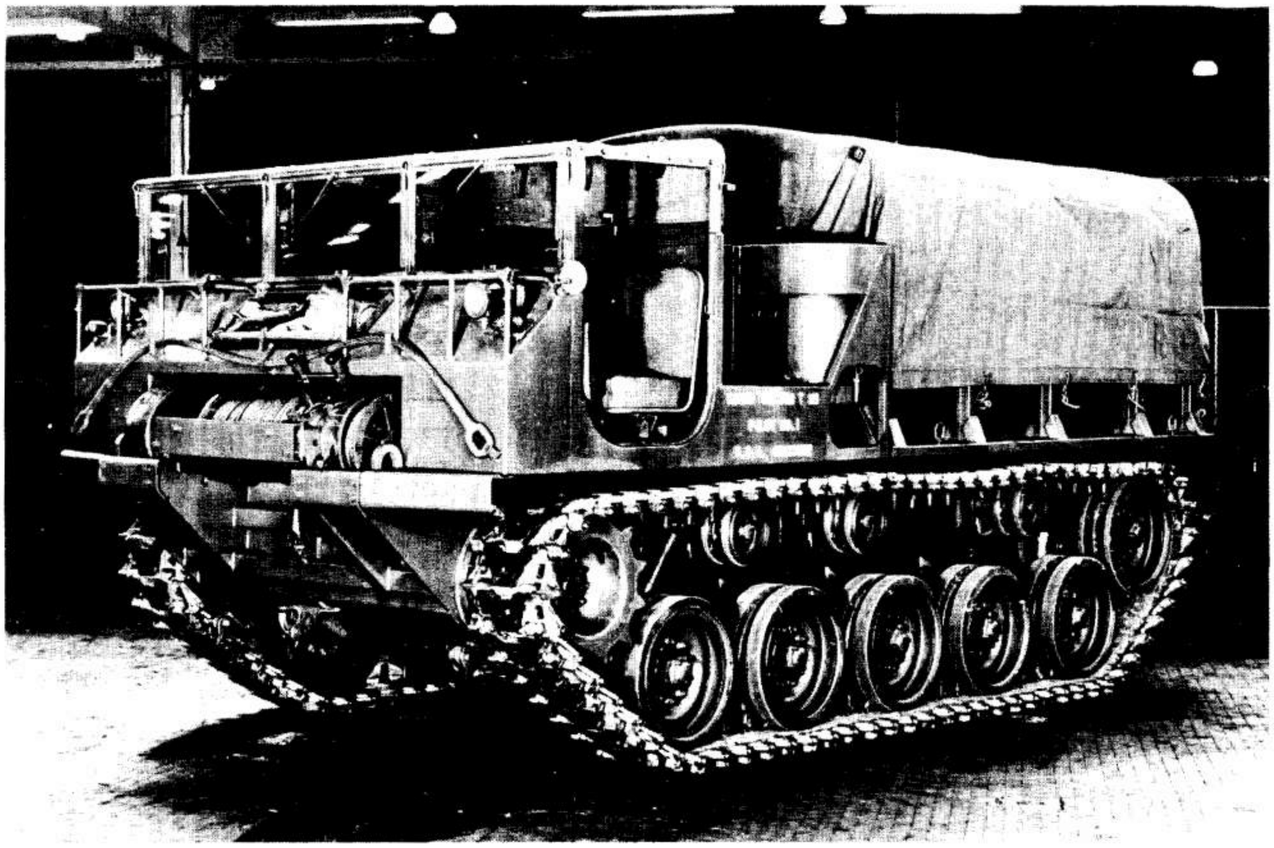


Figure 1-27. Cargo Tractor, 20-ton, M85

vehicle have to do with vision and protection to hydraulic lines. The vehicle driver and crew must have adequate vision from inside the vehicle to operate the various attachments without being exposed to enemy fire, and the hydraulic lines leading to the engineer attachments must be adequately protected from damage by projectiles and construction debris.

1-7.1.6 Tractors

Tractors, as the term is used by the military, designates all track-laying or wheeled motor vehicles designed for such functions as towing of artillery, trailers, sleds, and other vehicles; earth moving by means of attached dozer blades, buckets, or scrapers; or for push or pull operation of construction and related equipment. To increase their versatility, they are often provided with cargo platforms or cargo bodies, foldaway troop seats, fully enclosed cabs, ammunition racks, winches, power takeoffs, and defensive armaments. Wheeled tractors are not to be confused with truck tractors which are a separate and distinct family of vehicles. Truck tractors are discussed in par. 1-3.2.

The general designs of tractor hulls vary widely among the various models. Most designs contain a frame consisting of two longitudinal heavy-sectioned steel channels that are rigidly braced and attached to two crossmembers which extend on each side. The cargo bodies and cabs are fastened to this frame. Some hull constructions, however, are essentially frameless, being fabricated completely of welded plates. Doors and access covers are provided for the passage of crew and passengers, for access to cranes and winches, and for the performance of maintenance and service to the vehicle. Light armor is sometimes added to protect the crew and vital parts of the vehicle; however, the majority of tractor models are unarmored. One type of cargo tractor is shown in Fig. 1-27.

A somewhat unique tractor is the Universal Engineer Tractor shown in Fig. 1-28. This vehicle serves as a bulldozer, scraper, rough grader, dump truck, prime mover, and cargo or

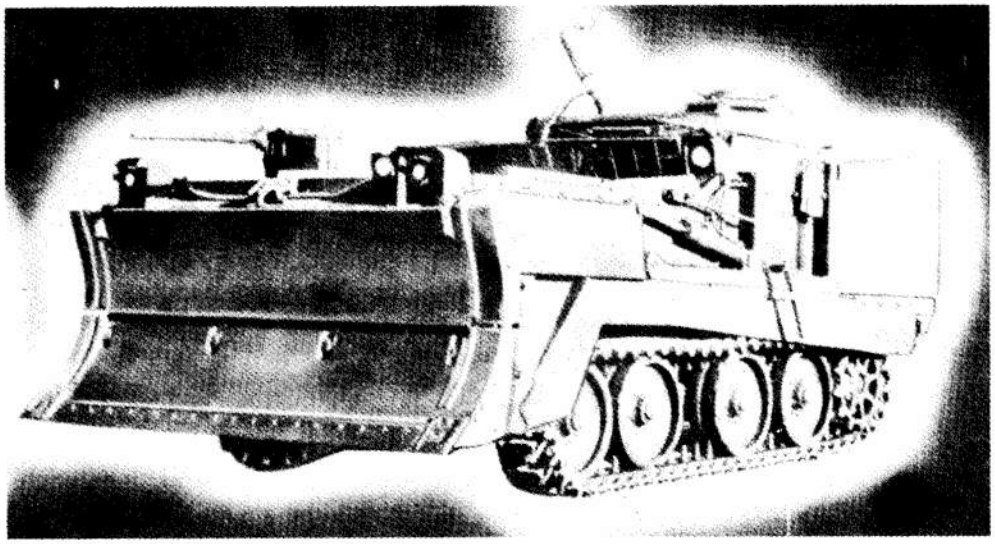


Figure 1-28. Universal Engineer Tractor, 12-ton, UET-E3

personnel carrier. It is self-loading and ballastable to increase its work output. In addition, it is air transportable, air droppable, amphibious, and provides light armor protection to its crew and passengers. The hull is of unitized welded aluminum construction with steel used for cutting edges and certain other components. Aluminum armor is used for lateral protection for the troops and all-around protection for the driver. The armor on this vehicle weighs about 4,000 lb, while the gross weight of the vehicle is 28,500 lb. It can carry a payload of 24,000 lb which gives it a payload-to-weight ratio of 0.84—quite high in comparison to other vehicles.

For descriptions and illustrations of other tractor types, both armored and unarmored, see Refs. 6 to 11, inclusive.

1-7.1.7 Special and Miscellaneous Vehicles

Special and miscellaneous vehicles are generally modifications of current standard vehicles; although, sometimes they may be specially designed for a specific, limited mission. This broad group of vehicles includes such items as flame-throwing vehicles, missile equipment carriers, missile loader-transporters, and mobile command posts and communications centers.

Flame-throwing vehicles are of two basic types: those on which the flame-throwing weapon serves as an auxiliary to the vehicle's main armament, and those on which it is the principal armament and with which the vehicle is integrally designed. In neither case is a new or different body design involved. In the latter case, a standard combat vehicle is modified to fire the flame gun through a specially designed dummy tube that resembles the vehicle's standard armament. This is done so as not to alter the silhouette of the unmodified vehicle, for a fundamental design doctrine governing flame-throwing vehicles is that they should bear no distinctive features by which they can be readily identified as flame-throwers. Since the flame-throwing vehicles utilize standard vehicles, no unique body designs are involved.

One important consideration associated with flame-throwers in general is the intense heat generated by the spewing flame. This may have undesirable and even hazardous effects upon the vehicle or some of its components, particularly when the flame is ejected against the prevailing wind. There is, however, a current model Flame Thrower, M132A1, which consists of an Armored Personnel Carrier, M113A1, modified by the addition of flame-throwing equipment.

Generally speaking, the special vehicles are adaptations of standard vehicles and include only such modifications to their hulls as are necessary for the mounting, operation, and servicing of any special equipment needed for the functional mission of the vehicle. Thus, the

XM462E2 Missile Equipment Carrier, the XM577 Light Command Post Carrier, and others all are based upon the M113 Armored Personnel Carrier (Fig. 1-25). Their names are sufficiently descriptive to indicate their functional purpose. In other adaptations this same basic vehicle is equipped as a mobile communications facility and as a mobile medical treatment station. Their hulls, however, are basically the same.

1-7.2 UNARMORED HULLS

1-7.2.1 Self-propelled Guns

There are, in general, two types of unarmored self-propelled guns. The first type, such as the M56, 90 mm Self-propelled Gun (Fig. 1-29), is unarmored in order to achieve the weight restriction imposed by phase I airborne operations. These weapon systems provide close support and antitank capabilities to airborne operations. Although these vehicles are subjected to direct enemy fire, they rely on their speed and agility for their safety. As the airborne weight limitations are relaxed, due to improved aircraft, it is probable that future airborne assault weapons will be lightly armored. This type of vehicle hull will be required to have swimming capabilities as well as the capability to withstand the weapon firing loads.

The second type of unarmored self-propelled guns is large caliber weapon systems employed primarily for counterbattery fire, the destruction of field works and reinforced concrete, interdiction fire, and to demoralize the enemy. This class of vehicles includes the 175 mm (Fig. 1-30), and 8-inch howitzers. These vehicles do not require extensive armor, since they are normally employed far enough behind the battle line to encounter only long range predicted fire weapon attacks. Furthermore,

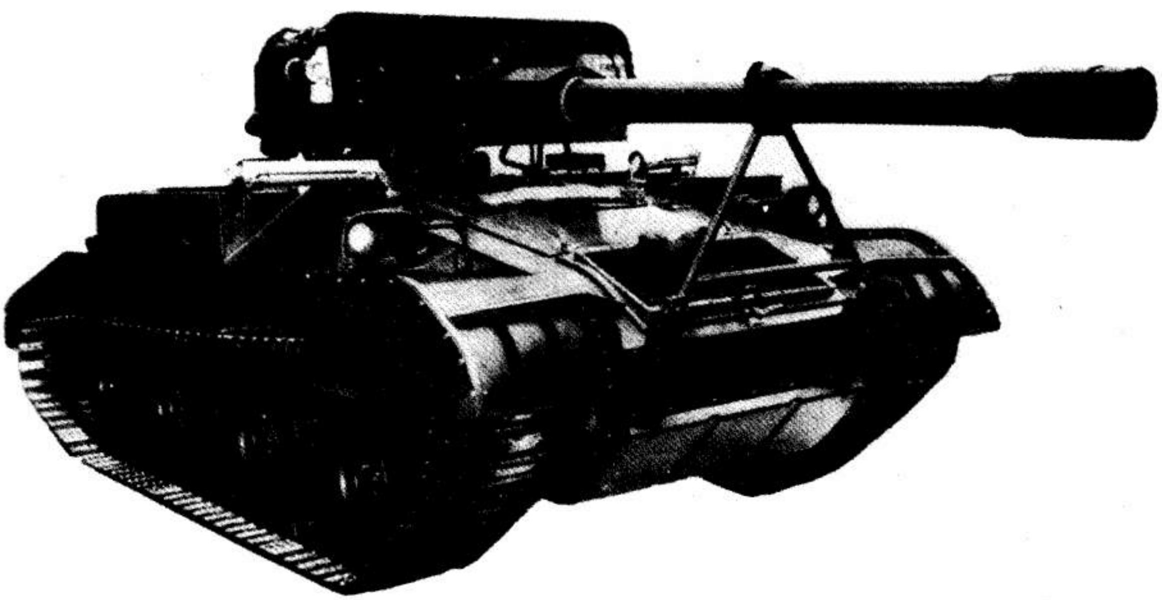


Figure 1-29. 90 mm Self-propelled Gun, M56

Figure 1-30. 175 mm Self-propelled Gun, M107

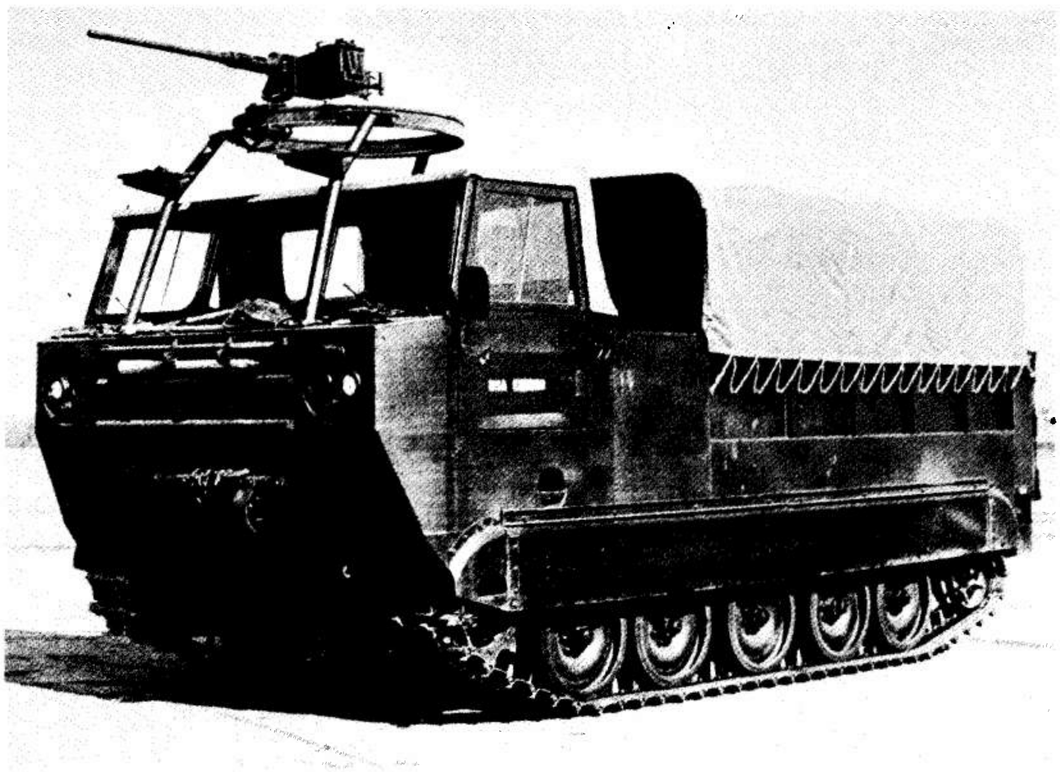


Figure 1-31. Track-laying Cargo Carrier, M548

since the weapon weight for this type of vehicle is large, the addition of armor would seriously degrade their mobility. The prime hull design consideration for this type of vehicle is the weapon firing reaction load.

1-7.2.2 Personnel and Cargo Carriers

The unarmored tracked personnel and cargo carriers perform many of the same functions as their wheeled counterparts. Their generally increased mobility over wheeled carriers, however, allows them to readily fulfill the supply requirements of tracked combat vehicles. A typical tracked Cargo Carrier is the M548 (Fig. 1-31). This vehicle uses the power package, running gear, and suspension of the M113A1 Armored Personnel Carrier. However, since the M548 is not armored, it achieves a weight saving in the order of 5,000 lb over the M113A1, thus adding to its mobility and payload.

For cargo and troop movement in the arctic and subarctic regions, a special type of cargo carrier is required. It is required to have a high degree of mobility in deep snow and over marshes, swamps, muskeg, and tundra under extreme climatic conditions. Adverse terrain vehicles of the past were such vehicles as the M29 and M29C Cargo Carriers, the T107 1/2-ton Amphibious Cargo Carrier, the M76 and the M116 Amphibious Cargo Carriers, and others. Descriptions of these vehicles can be found in Refs. 6, 9, 10, and 11. The latest vehicle of this type is the XM571 Articulated Utility Carrier shown in Fig. 1-32. Its two hulls are made of honeycomb panels used above the sponsons. They are completely watertight with no doors or tailgates and are designed to withstand moderate poundings from boulders and stumps during tactical operations. These vehicles will be used for command, reconnaissance, liaison, courier and passenger service; for carrying ammunition, equipment, and supplies; for limited casualty evacuation; for laying wire; and for towing ski troops and sleds.

1-7.2.3 Unarmored Recovery Vehicles

The unarmored recovery vehicles are designed to perform many of the same functions as do the armored recovery vehicles (par. 1-7.1.4); and, thus, their hull constructions are similar.

Armor plate is eliminated primarily to reduce the weight of the vehicle. This reduced weight affects the power plant, transmission, and fuel storage requirements which, in turn, also affect the vehicle weight. Some unarmored Recovery Vehicles (M578) utilize the hull-chassis of standard self-propelled artillery vehicles.

1-7.2.4 Unarmored Tractors

Unarmored tractors are similar in their functions and appearance to the armored tractors discussed in par. 1-7.1.6. Their chief differences are their lighter weight and greater speed made possible by the elimination of the heavy armor plate. For more discussion of these vehicles see par. 1-7.1.6.

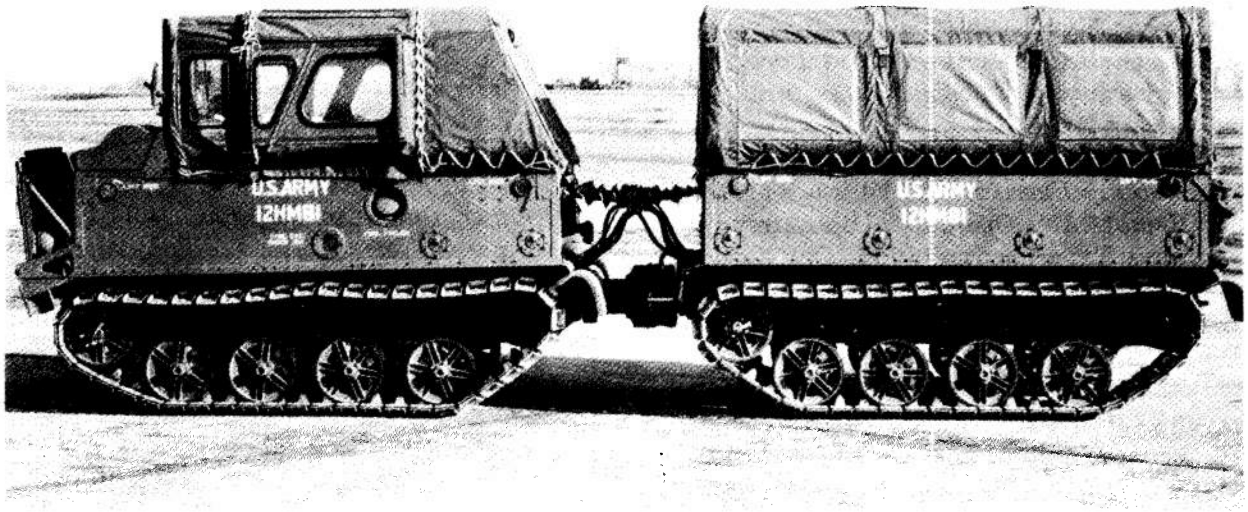


Figure 1-32. Articulated Utility Carrier, XM571

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CHAPTER 2
DESIGN CONSIDERATIONS*
SECTION I—LIMITING DESIGN FACTORS

The design of military vehicles is subject to various restrictions, requirements, specifications, and other controls which influence their dimensional, geometrical, and operational aspects. These limitations and controls arise from the requirements that military vehicles be compatible to unrestricted transport by road, rail, air, and seagoing vessels; from the need for tactical mobility under extremely adverse conditions of terrain and climate; from the need for simplified service and maintenance operations; and from certain theoretical and empirical military requirements which have evolved from past experience in the field. Detailed specifications and recommendations are given in various readily available military publications; therefore, their repetition here would be redundant. Furthermore, the specifications given in the original, official publications are revised and upgraded from time to time in keeping with current doctrine. Thus, when referring to these documents, it is important that the latest issue or revision be used. The paragraphs which follow cite the reference documents pertinent to each topic, give a cursory summary of their salient points, and discuss briefly how each relates to, or influences, the vehicle body or hull.

2-1 PHYSICAL LIMITS

2-1.1 GENERAL

Physical limits are the maximum values of size, weight, and weight distribution recommended for unrestricted conveyance of the vehicles by appropriate modes of transport in any country of the world. Limits are not given specifically for vehicle bodies or hulls; but, since they are the largest single element of most vehicles, their dimensions are the most significant to the overall dimensions of the vehicle. Dimensions and weights may exceed the recommended limits when it has been determined that transport of the vehicle will be limited to only certain transportation modes. In these cases, dimensions and weights of an item need only be adjusted to suit the capabilities of the specific transportation modes to be used.

The policy of the Department of Defense with regard to the transportability of materiel is given in AR 705-8, *Department of Defense Engineering for Transportability Program*[1]. It directs that major emphasis be given to transportability when assigning priorities for consideration during the design of new items of materiel. It is highly desirable that all materiel and equipment developed for use by the military be of such outside dimensions and gross weight as will permit ready handling and movement by all available transport facilities. In general, this resolves to an overall maximum length of 32 ft, maximum width and height of 8 ft, and a gross weight not to exceed 11,200 lb. Theoretically, designs that exceed any of these limitations will require special arrangements of schedules, rights of way, clearances, or other special handling procedures. Permission to exceed the recommended limits can be obtained from appropriate Transportability Agencies. These are listed in Ref. 1 along with the correct requesting procedure.

2-1.2 HIGHWAY TRANSPORTABILITY

Vehicles that are adaptable to movement by highway must be designed to meet the highway axle load limitations, highway and bridge loading allowances regarding gross vehicle or gross combination weight restrictions, size limitations, and statutory requirements for state certification as needed, whether or not the use planned for the vehicle envisions movement of the vehicle over public highways. Maximum allowances vary greatly among the states and are especially complex with respect to axle loadings and overall lengths of various combination rigs. Therefore, in all cases involving vehicle designs that exceed any one, or more, of the limits

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stated in par. 2-1.1, an evaluation and approval should be requested from the appropriate Transportability Agency designated in AR 705-8. A more comprehensive discussion of this subject is given in Refs. 2-8.

2-1.3 RAIL TRANSPORTABILITY

Physical limits imposed by railroads result from limiting dimensions and load capacities of railroad cars and by clearance dimensions along the rights of way. The dimensions and load capacities of railroad cars vary widely throughout each country of the world. Refs. 4-6 summarize the principal characteristics of some of the more commonly used cars on North American and foreign railroads. More detailed information on specific cars can be obtained from the *Car Builders' Cyclopedia*, Association of American Railroads, Simmons-Boeadman, N. Y., N. Y., and from *The Official Railway Equipment Register*, The Railway and Equipment Publishing Co., N. Y., N. Y., which list all of the cars used on American railroads. For detailed information on foreign railroad cars, consult *Jane's World Railways*[9], the applicable consulate or embassy, or the railroad industry of each country concerned.

Limiting dimensions imposed by clearances of railroad rights of way are summarized in the Outline Diagram of Approved Limited Clearances of the Association of American Railroads, for North American railroad clearances, and in the Berne International Clearance Diagram, for standard European railroad clearances. These are amply illustrated and discussed in Refs. 2-4. A more complete listing of railway right of way clearances, including that of railroads of Central and South America, Africa, Asia, and Australasia, is given in *Jane's World Railways 1965-66*.[9]

2-1.4 AIR TRANSPORTABILITY

When designing vehicles to be transported by aircraft, maximum effort should be made to enable them to be transported in the maximum number of types of aircraft available. Careful consideration must be given to the limitations imposed by the physical characteristics of the aircraft involved. The most important of these are:

a. Maximum allowable aircraft payload.

b. Size, location, and configuration of door openings.

c. Size and configuration of cargo compartments including limiting factors that may prevent the full utilization of available space.

d. Strength of aircraft floor and loading ramp.

e. Number, location, and strength of tiedown fittings.

f. Aircraft center of gravity limits forward and aft.

Other considerations are the limitations imposed upon the maximum length of the vehicle by the need for loading via an inclined ramp, and the effect cargo weight transfer has upon the effective door opening. As the vehicle moves up the inclined ramp to enter the aircraft, the upper portion of its forward end may strike the ceiling of the aircraft before the vehicle is completely on board thus effectively establishing a maximum allowable cargo length that may be shorter than the length of the cargo compartment. Loading from a horizontal platform made level with the cargo floor may not be an acceptable solution to this condition if the availability of such a horizontal platform for loading *and unloading* cannot be assured. The other consideration mentioned, that of cargo weight transfer, occurs as the weight of a heavy vehicle being loaded from a loading ramp is transferred from the ramp to the aircraft. This weight transfer produces a change in the attitude of the aircraft which modifies the effective door opening.

General requirements and considerations for air transportability are given in AR 705-35[10] and MIL-A-8421B[11]. Dimensions of cargo compartments and cargo doors for various commonly used commercial and military aircraft are summarized in Refs. 2, 4, 5, and 6. Specific loading and performance problems should be resolved by referring to the Technical Order applicable to the aircraft in question (e.g., Ref. 12). Additional data, recommended practices, and other useful guidance are given in Refs. 13-15.

2-1.5 SEA TRANSPORTABILITY

Size and weight limits imposed by requirements of sea transportability are governed by the dimensions of cargo hatches, heights of cargo compartments, load-bearing

characteristics of cargo decks, and load capacities of available cargo booms and winches. These limitations vary widely from ship to ship and even vary in different quarters of the same ship. In general, however, for storage on board most merchant ships, overall dimensions of the cargo package should not exceed 35 ft in length, 20 ft in width, and 11 ft 4 in. in height. The heights of 'tween deck holds (between the weather or main deck and the lower hold) range, among most ships, from 6 to 12 ft, while the height of the lower holds varies from 7 to 25 ft. It is obvious, therefore, that reference must be made to documents that give statistics and characteristics of specific merchant ships when exact information is required. These data are given in Refs. 4, 6, 16, and 17.

2-2 OPERATIONAL LIMITS

The operational requirements of military vehicles are specified in the Military Characteristics (MC's) pertaining to each vehicle. Based upon past experiences with vehicles in all parts of the world under every conceivable environment in peacetime maneuvers as well as during wartime operations, a number of these requirements have been standardized and made applicable to various types of combat and tactical vehicles. These are summarized in MIL-HDBK-134A[18] and discussed in Refs. 19 and 20. In general, these specifications are pertinent to the overall vehicle rather than applying expressly to the bodies or hulls. Body and hull design, however, can influence the attainment of these requirements and is, in turn, affected by the specified operational requirements. Some of these considerations and relationships are pointed out in the paragraphs which follow.

2-2.1 STANDARD OBSTACLES

The standard obstacles encountered by military vehicles are such things as bridges, steep slopes, vertical walls, ditches, trenches, washboard roads, staggered sideslopes (frame twisters), and water barriers.

2-2.1.1 Bridges

Bridges are obstacles when they cannot support the vehicle weight or when they are too narrow or lack sufficient headroom for passage of the vehicle. Compliance with specifications on physical limits discussed in the preceding paragraphs will reduce this possibility to a practical minimum.

2-2.1.2 Steep Slopes

Steep slopes can impede or arrest the forward motion of a vehicle in any one, or combination, of four ways; namely, (1) through loss of tractive effort below the value needed to climb the slope, (2) through toppling over of the vehicle, (3) through interference of structural components at the front or rear of the vehicle with the slope or ground, and (4) due to the vehicle bridging the angle at the transition from level ground to the slope. These are discussed to some length in Ref. 20.

The maximum tractive effort that can be developed by a vehicle depends upon the component of the vehicle weight that acts normal to the ground, upon the uniformity of load distribution upon the soil, and upon the coefficient of friction or adhesion that exists at the soil-vehicle interface. The weight component acting normal to the slope decreases in magnitude as the slope increases, the load distribution upon the soil shifts downhill and the nonuniform soil loading worsens as the slope increases, and the coefficient of adhesion at the soil-vehicle interface may degrade considerably if the soil becomes loaded beyond its capacity. These conditions all add up to a reduced tractive effort potential when on a slope which may be sufficient to stop the vehicle.

The possibility of tipping—in the lateral as well as the longitudinal direction—is increased on steep slopes due to the shifting of forces acting on the vehicle due to the pull of gravity with respect to its base. Dynamic forces due to vehicle acceleration and cornering can act to aggravate the situation. This consideration assumes special importance when the vehicle's center of gravity is high or when the vehicle is designed to carry a load outside of its wheelbase as in the case of crane-type vehicles, recovery vehicles, loaders, scoops, and shovels.

Obviously, the key factor in controlling loss of traction and tipping due to slope operations is the location of the center of gravity. Since most vehicles can be expected to travel uphill as well as downhill and on both right- and left-hand

sideslopes, there is little that the designer can do in the way of countering the effect of the shifting forces caused by the inclines by such measures as locating the center of gravity radically offcenter of the vehicle. The best compromise, then, is to locate the center of gravity laterally and longitudinally at the approximate center of the vehicle. In height, however, there is considerable advantage in locating the center of gravity as low as possible so as to minimize the overturning moment.

Interference of structural components of the vehicle with the slope on ground can occur at either end of the vehicle and at the bottom of either an upgrade or· a downgrade. This interference is a matter of insufficient angle of approach or angle of departure designed into the vehicle. These terms are defined in the Glossary at the end of this handbook and discussed in detail in Ref. 20.

The condition of a vehicle bridging the angle at the bottom of a steep grade and becoming immobilized is partly due to insufficient angles of approach and departure, partly due to the vehicle's suspension characteristics, and partly to the soil characteristics at the site. Suspension and soil characteristics are beyond the control of the body or hull designer, but the angles of approach and departure are not. The application of an articulated body structure to a long vehicle reduces the possibility of bridging.

2-2.1.3 Vertical Walls

The ability of a vehicle to climb vertical walls depends upon many factors, not all of which are in the domain of the body or hull designer[20,21]. Those which are in his domain are the angles of approach and departure and the location of the center of gravity. The former must be adequate to prevent interference from structural parts of the vehicle, and the latter should be centrally located and as low as possible.

2-2.1.4 Ditches and Trenches

Ditches and trenches are similar in that they are both long, narrow defiles in the ground. As the terms are used here, the ditch is considerably wider at the top than the length of the vehicle, generally has sloping sides that meet in a rounded "V" at the bottom, and its crossing requires that the vehicle enter it on one side, traverse the near side to the bottom or until it engages the opposite side, and then climb up the opposite side. Ditch-crossing capabilities depend upon such factors as slope-climbing ability, maximum angles of approach and departure, suspension characteristics, and the presence or absence of projections and overhangs on the vehicle that might interfere with the ditch profile. The discussion of bridging given in par. 2-2.1.2 applies even more to ditch crossing than to slope climbing because the angles formed by ditch walls are usually more acute than those formed by slopes.

Trenches are usually considered relatively narrow with respect to the vehicle length and have vertical or near vertical walls. In crossing trenches, the vehicle makes use of its length, longitudinal rigidity, location of its center of gravity, and its tractive ability to span the gap without entering it. Wheeled vehicles are generally not suited to trench crossing, since their suspensions do not permit cantelevering a portion of the vehicle over a gap. Tracked vehicles are much better suited for this purpose and can usually cross trench widths of about one-third the vehicle length.

While trench- and ditch-crossing capabilities of a vehicle are largely in the domain of the suspension system designer, the body or hull designer can contribute by helping to keep the vehicle center of gravity as low as possible, by providing a rigid structure, by eliminating troublesome overhangs and protrusions that may interfere with the ground, and by providing maximum angles of approach and departure.

2-2.1.5 Washboard Roads and Staggered Sideslopes

Operations over washboard roads produce repeated dynamic loads on the vehicle body or hull which lead to failure of structural members, mounting brackets, and piping due to material fatigue. The vibrations also affect threaded connections causing them to loosen or break, they tend to cause gasketed or sealed joints to leak, and they can cause failure in electrical connections and wiring. Noise is another problem which results when operating on washboard roads. This is encountered especially in lightweight vehicles. The thin body panels used are very prone to resonate and amplify the

noise, sometimes to an intolerable level. By supporting thin body panels to prevent or minimize their vibrating, by using stiffeners or stiffening corrugations, and by applying vibration isolating and damping materials, the noise problem can be controlled satisfactorily.

Staggered sideslopes (frame twisters) impose severe twisting loads upon body and hull structures. Wheeled vehicles are usually designed with a semiflexible frame and body which permit the structure to flex and deform elastically under these loads. Designs aimed at making these vehicles perfectly rigid under this type of service become unacceptably heavy. An exception is made in the case of cranes, shovels, wreckers, and similar vehicles where a certain amount of rigidity is necessary for the efficient functioning of the equipment. Even here, however, a compromise is made and a certain amount of elastic deformation is tolerated. Tank hulls and the hulls of similar heavy vehicles, on the other hand, are inherently so rigid that frame twister courses have little effect upon them.

2-2.1.6 Water Barriers

Water barriers comprise any land area that is inundated with water to a depth sufficient to impede or stop the movement of troops or materiel. This includes such features as rivers, ponds, lakes, bays, and oceans. Water-crossing requirements for future military vehicles are given in AR 705-2300-8[22]. The AR defines four general methods vehicles may employ in crossing water barriers, namely,

(1) *shallow-fording*. The ability of a vehicle, with its suspension in contact with the ground, to negotiate a water obstacle of a specified depth without the use of special waterproofing kits.

(2) *deep-fording*. The ability of a vehicle, with its suspension in contact with the ground, to negotiate a water obstacle of a specified depth by the application of a special waterproofing kit.

(3) *floating*. The ability of a vehicle to negotiate water obstacles without being in contact with the bottom. Self-propulsion while in the water is not implied in this definition.

(4) *swimming*. The ability of a vehicle to negotiate a water obstacle by propelling itself across without being in contact with the bottom.

As a minimum requirement, all combat and tactical vehicles should have an inherent capability of shallow-fording in fresh and salt water to the maximum depth practicable but not less than 30 in., including sinkage depth and wave height, with the following exceptions:

a. Tactical vehicles up to, and including, 1½-ton payload capacity must be capable of shallow-fording depths of not less than 20 in.

b. Tanks and other armored vehicles must be capable of shallow-fording depths of not less than 42 in.

Swimming or floating are the preferred modes of water-crossing for combat and tactical vehicles, and deep-fording kits are mandatory equipment for all vehicles of this type that do not have this capability. These kits are required to provide deep-fording capability for at least 15 min in fresh or salt water to the following depths (sinkage depth and wave height are included in the depths given):

a. Fully enclosed armored vehicles, to the maximum depth practicable consistent with adequate freeboard. Freeboard is measured from the top of the commander's hatch opening or turret.

b. All other vehicles, including trailers, 5 ft.

The impact of these specifications upon the design of vehicle bodies or hulls—such as watertight compartments, sealing of doors and other openings, drainage provisions, hydrostatic forces, buoyancy and stability, and other considerations—is discussed in other parts of this handbook.

2-2.2 CLIMATIC REQUIREMENTS

Military vehicles, in general, are required to operate in air temperatures ranging from -70° to 125°F and to withstand the effects of storage in ambient temperatures ranging from -70° to 160°F for specified time periods. Many vehicles, however, are not required to meet this entire temperature range, and use is often made of special climatization kits to extend the useful range of vehicles. The climatic environment for military materiel to be used in all parts of the world is specified in considerable detail in AR 70-38[23]. Operating conditions are divided into eight climatic categories, namely, wet-warm, wet-hot, humid-hot coastal desert, hot-dry, intermediate hot-dry, intermediate cold, cold,

and extreme cold.

Intermediate operating conditions are those which exist at certain times and places in the most densely populated portions of the world and where major military activities have taken place in the past. Standard general purpose military combat and combat support materiel is required to be capable of satisfactory performance at all times under these conditions. Recent experience has indicated that materiel should also be designed for use in the wet climatic categories. In brief the *intermediate* and *wet* conditions include the following:

a. Thermal stress—upper limit. Four hours with an ambient air temperature of 105°F with an extreme temperature of 110°F for not more than one hour, a maximum ground surface temperature of 130°F, solar radiation (horizontal surface) of 360 Btu/ft^2-hr, and a wind speed between 5 and 10 knots.

b. Thermal stress—lower limit. Six continuous hours with an ambient air temperature of -25°F, a ground or snow surface temperature of -35°F, and wind speed less than 10 knots.

c. Temperature-humidity stress.

(1) Relative humidity above 95% in association with temperatures near 75°F for a period of a day or more, simulating conditions found under the canopy of heavily forested tropical areas.

(2) Four hours with an ambient air temperature of 95°F, ground temperature and solar radiation same as in thermal stress-upper limit (par. a), relative humidity of 74%, dew point of 85°F, simulating conditions in the open in tropical areas.

d. Relative humidity—upper limit. 100% at all ambient air temperatures from -25° to 85°F.

e. Relative humidity—lower limit. 5% at 110°F ambient air temperature.

f. Precipitation. Wind-driven rain falling at specified rates for various time intervals to a maximum of 9.5 inches over a 12-hr period accompanied by intermittent winds to 35 knots and including short periods of torrential downpour as specified in AR 70-38.

g. Snowloads. Uniform loads of 10, 20, and 40 psf depending, respectively, upon whether the equipment is portable (moved daily), temporary (moved often and cleared of snow between storms), or semi-permanent (snow not usually removed between snowfalls).

h. Wind velocities. Speeds of 45 knots for a 5-min period with gusts to 65 knots and with hold-down or other facilities provided to withstand winds of 55 knots for a period of 5 min with gusts to 85 knots. Higher winds occur in exposed coastal and mountain locations.

i. Falling snow. Crystals of 0.05 to 20.0 mm in diameter with a median range of 2.0 to 5.0 mm.

j. Blowing snow. Particles occur in greater numbers in size range 0.02 to 0.4 mm diameter. Median particle size varies with temperature and wind speed with smaller sizes occurring at lower temperatures and stronger winds.

k. Blowing sand. Particles of 0.01 to 1.00 mm in diameter at 15 knots or more.

l. Blowing dust. Particles of 0.0001 to 0.01 mm in diameter blowing at 15 knots or greater.

The *hot-dry* category extends the thermal stress experienced by the equipment to a 4-hr period with an ambient air temperature of 120°F with an extreme temperature of 125°F for not more than 1 hour and a ground surface temperature of 145°F. The minimum relative humidity associated with this category is 5 per cent at 125°F.

The *humid-hot coastal desert* category extends the temperature-humidity stress to the joint occurrence of a temperature as high as 100°F with a dew point of 86°F.

The *cold* category extends the ambient air and ground surface lower temperature limit to -50°F for 6 hours. Wind speed is less than 10 knots.

The *extreme cold* category extends the ambient air and ground surface lower temperature limit to -70°F for 6 hours. Wind speed is less than 5 knots.

The last four climatic categories are less prevalent than the intermediate and wet categories. Therefore, for operations in hot-dry, humid-hot coastal desert, and cold conditions modification kits may be developed to extend the use of equipment designed for intermediate and wet conditions. Equipment will be designed specifically for the more extreme categories only if modification kits are impractical or if such specific designs achieve meaningful improvement. The last category, extreme cold, occurs only in a few places in North America, Greenland, Siberia, and Antarctica. Operations in these areas may require that equipment be

designed specifically for extreme cold conditions.

In addition to meeting the requirements of specified climatic categories, all military materiel must be capable of safe storage and transportation without permanent impairment of its capabilities by the effects of extreme environmental conditions. These conditions are grouped under three headings, and their salient features are summarized as follows:

High temperature storage. Air temperature up to 155°F for periods up to 4 hours daily, with a maximum temperature of 160°F for not more than 1 hour.

Low temperature storage. Air temperatures down to -70°F for 6 hours and temperatures below -60°F for 24 hours.

Air transit conditions. For materiel subject to shipment by air (elevations as high as 50,000 feet), an air temperature of -105° F and a pressure of 100 millibars are applicable. These conditions could result from failure of cabin pressure and temperature regulators.

This summary of climatic design criteria given here is only a superficial treatment. AR 70-38[23] is far more comprehensive and includes complete specifications—for all eight categories on thermal stress, water vapor (humidity and dew point), precipitation, snowloads, winds, atmospheric pressure, blowing snow, sand and dust, and more. The areas where the eight categories apply are shown in a map included with the AR. Consult the AR in all cases requiring more than a general knowledge of this subject. Additional and supplemental discussions on this subject can be found in Refs. 24 to 26.

2-2.3 AIR DELIVERY

2-2.3.1 General

The terms "air delivery" and "air transportability" as used by the military are not synonymous. Both terms deal with movement of personnel and materiel by air, but the main distinction of air delivery is that the cargo is unloaded while the aircraft is in flight. This imposes far more severe requirements upon the designer than those necessary for air transportability. From the time the air delivered item is placed on board the aircraft until it is recovered on the ground and placed into operation it will have been subjected to a loading environment produced by the following conditions:

a. Restraint of the item within the aircraft for reasons of flight safety.

b. Unloading of the aircraft in flight by parachute or other extraction device.

c. Deployment of the parachute recovery system.

d. Deceleration on ground impact.

Regulations governing the development of materiel for air delivery are given in AR 705-35[10].

2-2.3.2 Tiedown Provisions

In order to prevent displacement of the vehicle while in the aircraft, provisions must be made on the equipment for tying it down securely. Detailed requirements for these tiedowns are given in MIL-STD-814A[27]. Tiedown provisions should be designed for maximum working loads (applied in any direction below the horizontal) of 5,000, 10,000, or 20,000 lb in accordance with the item total weight being, respectively, under 5,000 lb, between 5,000 and 15,000 lb, or over 15,000 lb and with yield strengths equal to or greater than these limits. Their ultimate strengths should be at least 1.5 times the maximum working loads.

The total number of tiedowns required per side is computed from the following relationship:

$$N = \frac{1.1W}{K} \tag{2-1}$$

where

N = number of tiedowns per side
W = total weight of item, lb
K = 1250 for 5,000-lb tiedowns
= 2500 for 10,000-lb tiedowns
= 5000 for 20,000-lb tiedowns

In no case should less than four tiedowns be provided (total for item of equipment). Tiedown spacing should be approximately equal and arranged in a regular pattern by using corresponding points of attachment on opposite sides of the equipment. Where the width of the equipment exceeds 75 in., additional tiedown provisions may be required across the front and rear.

The inside openings or eyes of tiedown provisions should have a minimum diameter or clear opening of 3½ in. Bearing surfaces should have a minimum width of ½ in. and have a minimum radius of ¼ in. on all edges.

A minimum clearance of 3 in. should be provided around all tiedown provisions to facilitate ease of engagement. Tiedown provisions should be beveled, rounded, or otherwise relieved to prevent cutting or fraying of tiedown devices and to avoid stress concentrations.

2-2.3.3 Suspension Provisions

In air delivery, the equipment being airdropped experiences four separate shocks; the first is produced by the extraction system which accelerates the item out through the cargo hatch, and the last is produced on impact with the ground. The other two are produced by the deployment of the parachute recovery system and consist of the snatch force and the parachute opening shock. The snatch force arises from the comparatively rapid deceleration of the deploying parachute recovery system in relation to the slow deceleration of the suspended load. This difference in velocities must be reduced to zero, and hence the shock of the snatch force.

The snatch force occurs from 0.1 to 1.0 sec ahead of the opening shock and is often considered as part of it. Opening shocks, however, can be reduced considerably through the application of special reefing, venting, collapsing, and other techniques to the parachute design, but snatch forces are much more difficult to control. At low aircraft speeds, snatch forces do not exceed opening shocks; but at higher speeds and when modern canopy designs are used, snatch forces can be the limiting factor in air delivery operations. Methods for calculating snatch forces and opening shocks are given in Ref. 29.

Suspension provisions are required, integral with the item to be air-delivered, for attachment of the parachute recovery system. These suspension provisions must be adequate to withstand the snatch and opening shocks. Integral hub attachments on wheeled vehicles may be used if they meet the detailed requirements of MIL-STD-814A[27]. This document specifies that each suspension provision should have a maximum working load capacity acting in any direction above the horizontal plane of 1.5 times the total weight of the item and a yield strength 1.5 times the maximum working strength. The ultimate strength of each suspension provision should be 1.65 times the maximum working load. A total of four such suspension provisions is required. These should be located as far apart as is practicable; and, if possible, in a rectangular pattern about the vertical axis through the center of gravity and at such a height above the horizontal plane passing through the center of gravity that interference between the suspension slings and the equipment will be minimized. Details on limiting dimensions of suspension provisions can be found in MIL-STD-814A [27].

2-2.3.4 Extraction Provisions

Extraction provisions are designed to provide for attachment of the parachute extraction system. Standard vehicle pintles or tow bar attachments can be used for this purpose provided they meet the requirements given in MIL-STD-814A[27] and the air-dropped item has a gross rigged weight less than 25,000 lb. The maximum working load capacity of extraction provisions should be 1.5 times the weight of the equipment, and their yield strengths should be 1.5 times their maximum working strengths. Ultimate load capacities of extraction provisions should be a minimum of 1.65 times the maximum working load for gross weights less than 25,000 lb, and 1.75 times the maximum working load for heavier items. The extraction provisions should be located on the longitudinal centerline of the item and below its center of gravity.

2-2.3.5 Ground Impact

The most severe loading experienced by equipment during air delivery is the landing shock. Ref. 30 gives shock values of from 40 to 100 g's experienced with air delivery equipment in use at the present time. New developments have as their goal the reduction of this shock to 8 to 15 g's. Design guidance for the development of equipment to survive landing shocks is given in MIL-STD-669[31].

In order to provide the maximum protection to materiel against damage on ground impact, the equipment should be designed to accommodate energy dissipators. Energy dissipators currently in use are paper honeycomb, class e, style A of MIL-H-9884, commercially designated as 80-0-½, expanded doublefaced, 3-inch thick panel. This

dissipator crushes at an essentially constant dynamic stress of 6,000 psf (±10 percent) through 0 to 70 percent strain. The crushing stress rises rapidly beyond 70 percent strain.

Equipment designed to be air-dropped should be capable of withstanding a deceleration force level of 19.5 (±10 percent) times its weight when decelerated from a downward velocity of approximately 25 fps to 0 fps on ground impact.

The decelerating force level F_D in lb can be calculated from the following simple expression:

$$F_D = W(G + 1) \tag{2-2}$$

where

W = weight of item, lb
G = deceleration rate in g's where
$g = 32.2$ ft/sec^2

The area of dissipator required to develop a specific deceleration force level is

$$A_D = \frac{W(G+1)}{S_a} \tag{2-3}$$

where

A_D = dissipator area, ft^2
S_a = average dynamic crushing stress of dissipator, psf

The relationship among impact velocity, deceleration force level, and dissipator characteristics is

$$V = \sqrt{2gGEt} \tag{2-4}$$

where

V = impact velocity, fps
g = acceleration due to gravity, 32.2 ft/sec^2
E = dissipator thickness efficiency = 0.7
t = thickness of dissipator, ft

A minimum final design value of 18.5 has been selected for G based upon investigations of standard military vehicles under drop tests conducted in accordance with MIL-STD-669. All items that passed the tests were successfully air delivered.

SECTION II—GENERAL CONSIDERATIONS

2-3 MAINTENANCE[32-34]

2-3.1 CATEGORIES OF MAINTENANCE

The maintenance concept approved by the U.S. Army is given in AR 750-1[32]. The term maintenance, as it is used by the Armed Forces, is all action taken to retain materiel in a serviceable condition or to restore materiel to serviceability. This includes such actions as inspecting, testing, servicing, classifying as to serviceability, repairing, overhauling, rebuilding, modifying, and modernizing. Maintenance is divided into four major categories; namely, organizational, direct support, general support, and depot maintenance. These divisions result in maintenance strata that relate the difficulty of the maintenance tasks to the skills of the personnel performing the maintenance and to the facilities available at each level.

Organizational maintenance is that maintenance which is authorized for, performed by, and the responsibility of a using organization on its own equipment. Normally, this maintenance consists of inspecting, cleaning, servicing, preserving, lubricating, adjusting, and the replacement of minor parts that does not require highly technical skills. It encompasses that degree of maintenance which is performed by the equipment operator or the operating crew, as prescribed by pertinent technical publications, or that can be performed by specially trained personnel in the using organization (organization's mechanics and technicians).

Direct support maintenance (formerly designated as third echelon of field maintenance) is that maintenance which is authorized for and performed by designated maintenance activities and maintenance units in direct support of the using organizations. Normally, this category of maintenance is limited to the replacement of unserviceable parts or subassemblies in order to restore an end item (vehicle, etc.) to serviceability on a return-to-user basis.

General support maintenance (formerly designated as fourth echelon of field

maintenance) is that maintenance which is authorized for and performed by units that are organized as permanent or semipermanent shops to serve lower echelons of maintenance.

Normally, these units overhaul or repair materiel to required maintenance standards in a ready-to-issue condition.

Depot maintenance is that maintenance which is required for the repair of materiel that requires a major overhaul or complete rebuilding of parts, subassemblies, assemblies, or the end item as required. Additional information on maintenance responsibilities, shop operations, definition of maintenance terminology, and maintenance criteria is given in Refs. 32-37.

2-3.2 MAINTENANCE CRITERIA

In January 1959, the U. S. Army Tank-Automotive Center (now Command) issued an official document entitled *The Maintenance Criteria for Ground Vehicles* which established certain maintenance objectives that were to be the design goals for all agencies engaged in the design and development of military vehicles. Since that time, these objectives have been approved by the Department of Defense and incorporated in MIL-STD-1228, *Maintainability Criteria for Tank-Automotive Material*[33] for mandatory use by the Departments of the Army, Navy, and Air Force effective 27 September 1962. These maintenance criteria are as follows: "The Design Facility shall have as [its] maintenance goal that ground vehicles will accomplish the following in a military environment, unless otherwise directed:

(a) "Wheeled, tactical vehicles: 25,000 miles without [field] or depot maintenance

(b) "Tracked vehicles: 5,000 miles without [field] or depot maintenance.

"*Maintenance Skill Level.* The Design Facility shall have, as a maintenance goal, that the total scheduled and unscheduled maintenance manhours shall be a constant ratio and not to exceed the following, within the requirements of a or b of the foregoing:

(a) "Self-Propelled Wheeled Vehicles. Seven percent of the operational hours. The average distance negotiated by the vehicle shall be considered to be 20 miles for each hour of operation.

(b) "Self-Propelled Tracked Vehicles. Twenty percent of the operational hours. The average distance negotiated by the vehicle shall be considered to be 10 miles for each hour of operation."

2-3.3 DESIGN CONSIDERATIONS[34-37]

Proper maintenance of mechanical equipment is required to achieve maximum service life. This is especially true of military automotive vehicles, which must be always ready to function under severe conditions. The more complex a mechanism, the greater the number of maintenance operations that are usually required. Modern military automotive equipment, although made initially rugged and reliable, cannot endure for long without adequate preventive maintenance. The Army Maintenance System is the end result of a long chain of logistic functions, and neglect during design and development may render maintenance extremely difficult and even impossible.

Military equipment should be so designed as to be capable of maintenance during severe military usage by means of readily available skills, tools, and supplies. The objective of maintenance engineering is to reduce the time during which equipment is denied to the user, and also to reduce the manpower, tools, equipment, and supplies that are required to perform competent maintenance. These objectives can be more readily attained if sufficient consideration is given during the equipment design phase to some of the following factors:

a. Components should be so designed and installed as to provide adequate working clearances and visibility for ease of servicing, adjusting, removal, and installation.

b. Service points for checking, replenishing, and draining of fuel, lubricant, hydraulic fluid, pneumatic pressures (including tires), coolant, electrolyte, etc., should be readily accessible and should incorporate features that facilitate these operations without being vulnerable to damage or contaminated.

c. Fuel tanks with capacities in excess of 50 gal should be capable of accepting fuel at the rate of 50 gal per min. Tanks having capacities of less than 50 gal should be capable of being filled within one min.

d. In general, all parts whose working surfaces are subject to wear or deterioration should be provided with appropriate means for lubrication. Exceptions to this are certain surfaces, such as

tank-drive sprockets, on which lubricants are objectionable. Permanently lubricated assemblies and assemblies that require no lubrication should be used at all points where they can meet the rigorous requirements of the military environment and where their application is economically feasible. Porous, lubricant-impregnated bearings and rubber-bushed journals are examples of such devices.

e. Materials should be resistant to or protected against chemical and electrolytic corrosion brought about by atmospheric conditions and galvanic action between dissimilar metals in contact, and against normal wear and abrasion to the extent that such deterioration will not reduce the effectiveness of the equipment nor appreciably increase its maintenance requirements. Particular attention should be given to surfaces subject to wear and abrasion, such as running boards, cabs, floor boards, and load decks, and to small, light parts that are vulnerable to corrosion, such as sheetmetal items, screws, nuts, bolts, springs, retaining chains, and other thin-gage parts.

f. All electrical, pneumatic, hydraulic, and fuel systems should be resistant to corrosion and fungi and be protected against the entry of foreign matter. When selecting materials for these systems, careful consideration should be given to their compatibility with the fluids they will contain. Copper and high copper content alloys, for example, should not be used in contact with modern fuels. Copper has a catalytic effect on gasoline causing high gum deposits [35].

g. Exposed surfaces should be shaped to avoid recesses that tend to collect and retain dirt, water, cleaning fluids, servicing fluids that may have been spilled or lost during operation, or foreign matter. Where such recesses cannot be avoided, suitable deflectors, closures, and drains should be provided.

h. Equipment should be designed to require a minimum number of periodic maintenance adjustments. Those maintenance adjustments that cannot be eliminated should be simplified to permit their accomplishment at the lowest practicable maintenance level.

i. To expedite the replacement of components, suitable aligning, piloting, guiding, lifting, and positioning features should be incorporated into the design.

j. The least possible number, sizes, and types of fastening devices should be used to minimize the number of operations and tools required for the removal and installation of components.

k. Maintenance operations should be capable of being performed by personnel wearing arctic clothing, including heavy gloves, to the maximum extent practicable.

l. Equipment should be designed to permit maintenance operations within a reasonable time after halting the vehicle, without danger to the maintenance personnel of being burned. Components which must be handled under these conditions, particularly heavy items that require a long time to cool, should be provided with handles, eyes, or other suitable devices.

m. The design of equipment should, to the maximum extent possible, permit maintenance adjustments with the standard tools issued with the vehicle.

n. Emphasis should be placed upon the use of a minimum number of line items of supply in maintenance, the use of standardized parts and hardware that are, in general, used throughout the military supply system [36], and the use of interchangeable parts and assemblies, particularly those incorporated into other equipment supported concurrently by direct support maintenance units.

2-4 RELIABILITY [38-41]

2-4.1 DEFINITION

The term "reliability" is generally defined as the probability that a device will perform a specific function at, or above, a desired performance level under prescribed environmental conditions for a specified period. By being based upon a probability factor, the definition recognizes that 100 percent reliability is unattainable; and since the definition is concerned with the maintenance of specified levels of performance, the measure of reliability is the determination of the percentage of the total units considered that meet the performance requirements. The probability of equipment meeting its performance requirements can be predicted by appropriately summing the failure rates of each component part. These failure rates are determined by the application of statistical methods. Many

references exist on these subjects; Refs. 38 through 40 are three examples recommended for background information on the application of these techniques.

The attainment of a high degree of reliability is largely a function of good design and of the designer's recognition of the need, importance, and the determining factors of reliability. To achieve this end, the designer must ascertain, as early and as realistically as possible, the specifications covering his final product, the environmental conditions and their influence upon the performance of his product, the human engineering aspects relating to his product, emergency conditions likely to be encountered, the types of failure to which his various components may be subjected, and the functional dependency existing among the various components of which his product is comprised. Having ascertained these factors, he has a frame of reference against which to evaluate his design.

2-4.2 TYPES OF FAILURE

In order to appreciate the element of uncertainty associated with reliability, one must have some understanding of the general nature of equipment, or component, failure. In general, there are three types of failure which have been classified[41]—wear-out failure, constant-hazard-rate failure, and infant mortality failure.

Wear-out failure is associated with the gradual loss of material, strength, or efficiency as from rusting or wear, fatigue of components, or degradation of performance characteristics due to aging or environmental effects. It is a progressive reduction of product capability until the product degrades to a point of physical failure or its functional performance falls below an acceptable level. This type of failure is relatively simple to predict after a number of life tests are performed.

Constant-hazard-rate failures are a type that is inherently unpredictable and are, therefore, likely to occur at any time during the life of the product. A chip of steel falling into a gearbox to ruin the gear teeth, a hidden defect in a casting that results in an early failure of a component, or a tire running over an unexpected spike are examples of this type of failure. Since such failures are apt to occur at any time, they are assumed to be evenly distributed during the life of the item.

Infant mortality failures, the third type, are fairly well known. They include the types of failure that occur at the time, or very soon after, a newly assembled device is first placed in operation. These types of failures are generally due to faulty assembly, fabrication errors, or design errors. Fortunately, failures of this type usually take place during the initial testing or running-in phase of the development.

2-4.3 FUNCTIONAL DEPENDENCY

The subject of reliability is sufficiently complex to warrant a complete series of handbooks on that subject alone[113]; therefore, any presentation of such a broad subject in only a few pages must be extremely simplified and general in nature. Functional dependency, however, is an important consideration in most reliability analyses and, as such, warrants a brief discussion here.

Functional dependency refers to devices or systems composed of several components whose individual functioning depends upon the proper functioning of preceding components—in other words, a series arrangement. The ignition circuit of an internal combustion engine, for example, depends upon the sequential functioning of the battery, ammeter, switch, ignition coil, distributor, and spark plugs plus the electrical conductors that connect these components. A malfunction of any one of these will affect the engine performance and can result in complete failure of the engine. Similarly, the functioning of a dump truck body depends upon the sequential functioning of many elements—such as controls, power-take-off assembly, hydraulic pump unit, operating valve, hoisting cylinder, body hinges, end gate control linkage, hydraulic lines, body stiffeners—each of which is a complex assembly in itself whose reliability is also subject to the functional dependency of its components. A failure of any one of these will affect the performance of the dump body. The reliability of such functionally interdependent sytems depends upon the interactions of the reliabilities of their individual components and the probabilities that each will perform satisfactorily for the required period. The overall reliability of the system, then, is the product of the individual reliabilities.

If the six major components of the ignition system just cited each had a reliability of 99 percent, the overall reliability of the system would be $(0.99)^6$ or 94 percent. If the system contains several identical components, any one of which can cause a failure, each must be included as a separate item in the computation of the system reliability. Thus, if we include six separate spark plugs, six spark plug leads, one high-tension lead, one low-tension lead, one battery grounding lead, and one lead from each battery to ammeter to switch to coil to distributor, the total number of components in the preceding example becomes 25. If each has a reliability of 99 percent, the system reliability is only $(0.99)^{25}$ or 77.8 percent.

It must be understood that this is a highly simplified illustration of the method used in determining system reliability. In an actual system the components have different failure characteristics. The probability distribution of wear-out failures will be different for each type, and the constant-hazard-rate failures of each type will have their own time characteristics. For more detailed procedures consult the references cited earlier.

2-4.4 IMPROVING RELIABILITY

Reliability improvement is primarily a matter of good design. Perhaps the most important single aspect of design, from a product reliability sense, is simplicity. Reliability, in general, increases with simplicity and decreases with complexity. In making a product simple, the designer must avoid designing multiple functions into an item; and he must strive to keep the parts that make up a system, as well as the interaction between parts, to an absolute minimum. Along with simplicity of design he must arrange for ease of accomplishing required service and maintenance tasks.

The judicious selection of materials and types of basic elements, such as castings or forgings in place of weldments or other composite structures, will usually result in increased reliability as well as in reduced cost. Along with the judicious selection of materials to improve reliability is the judicious selection of lubricants and seals. This is particularily important when the equipment will be subjected to harsh environments, as is usual with military equipment.

Redundancy is a practice often employed to improve reliability, especially in electronic systems. In general, redundancy involves the installation of more than one part, circuit, or subsystem in a parallel arrangement, to perform the same function. Thus, in a system where one component or subsystem has an inherent low reliability, the system reliability can be increased by installing two, or more, subsystems so that any one can keep the system operating satisfactorily. This practice is not restricted to electronic systems, however, and is evident even in the design of bodies and hulls for military vehicles. For example: four lifting eyes are required on a tank hull—each to be capable of supporting the total weight of the vehicle; more than one escape hatch is usually provided in a hull; more than a minimum number of vision ports are provided; and other examples can be cited.

Two forms of redundancy are employed; standby redundancy and continuous redundancy. *Standby redundancy* describes the use of duplicate units or subsystems, in a functionally parallel arrangement, that can be brought into operation automatically or manually when the primary item fails. *Continuous redundancy* describes the use of duplicate units or subsystems, in a functionally parallel arrangement, that are in continuous operation so that the system will continue to operate at an acceptable level despite the eventual failure of a primary item.

Redundancy should be employed with discretion because of its impact upon system complexity and its economic import. Redundancy will not increase reliability if the redundant components cannot withstand the operating environment. Furthermore, the necessity of resorting to redundancy is sometimes an admission of unsatisfactory reliability of a primary item and indicates a requirement for improvement of the deficient component. In such cases, the designer should strive for perfection in design rather than the increased complexity attendant to redundancy, and redundancy should be employed only temporarily until design perfection can be achieved.

2-5 VALUE ENGINEERING

2-5.1 GENERAL[42,43,47,48]

Value engineering may be defined as an objective appraisal of a system, assembly, or component to achieve a reduced overall cost without compromising function, performance, reliability, or durability. The component parts of value analysis have been known and applied, in some form, to industrial practices and production for years. A cost analysis has been used, sometimes with total disregard to engineering principles, to reduce item or assembly costs. A functional analysis has been used, in several forms, to determine whether an assembly can be made to perform the same function with fewer parts. Material and fabrication analyses have been widely used to provide the same component function at a reduced cost. Value engineering simply systematizes the various detailed analyses into an overall value analysis. Principally, value analysis combines cost analysis and function analysis. Material, durability, reliability, manufacturing and performance analyses may also be required either in the form of a complete analysis or simply by being aware of their requirements and making a "judgment" as to the extent each area is affected.

2-5.2 COST ANALYSIS[42]

A cost analysis is used to isolate detail cost factors relating to the manufacture of the assembly or the component. These costs include materials, labor, services, and overhead. The cost estimates must be made on an annual or production run basis (whichever is shorter) and must include scrap, rework, rates of rejection, etc. No value judgment is based on the cost analysis alone. The analysis does, however, highlight areas of possible savings.

2-5.3 FUNCTIONAL ANALYSIS[42]

Functional analysis is a technique that assigns a verb and a noun to each component and process which describe its primary function. A door hinge, for instance, could be described in the verb-noun relation as, "provides rotation". After each part of an assembly or process is described in this manner, the results are tabulated. Repetitive and redundant functions that would otherwise be missed in normal design and manufacturing practices will show up in the tabulation.

After each component function has been defined and tabulated, its cost is ascertained from the cost analysis. This cost, however, does not necessarily determine its worth. A function is worth its cost only if there is no acceptable alternate method of performing the function at less cost. The term "acceptable alternate" requires the investigation of the effects of component change on the other analysis areas such as materials, durability, reliability, and performance.

2-5.4 MATERIAL ANALYSIS

The purpose of the material analysis is to minimize the cost of the materials used in the component or assembly without degrading the remaining requirements. After the functional analysis is completed, a tabulation is made listing the components, their material, raw material size, method of manufacture (casting, forging, etc.), cost of materials, and manufacturing costs. An analysis of these parameters may indicate alternate material selections, alternate component shapes or sizes to take better advantage of standard stocks, and alternate methods of manufacture. As in the functional analysis, the material is worth its cost only if no acceptable alternate material can be obtained.

2-5.5 EFFECTS OF MANUFACTURING METHODS

The effects of manufacturing methods are important considerations in any complete value engineering analysis. Considerations should be given to areas such as tools and dies, jigs and fixtures, number of machining operations, rate of production, surface finishes required, and design tolerances. Alternate manufacturing methods, for instance, may allow a choice among forging, casting, machining, or welding, each of which has inherent characteristics.

In general, forging is a high production process in which the cost of tools and forging dies must be amortized over many pieces to be economical. Casting, on the other hand, is a low production, or slow production, rate process. While the casting process incurs mold costs,

these are usually low-unless the pieces are complex—and are amortized over relatively few pieces. Machining is used in both high-and low-rate production depending upon the part. Where intricate machining and good surface finishes are required, machining offers a reasonable production method. High machining production can be accomplished on automatic screw machines if the components are adaptive to this method. Welding is an attractive alternate to casting, at times, but it is limited to sections of similar thickness to minimize warpage.

The selection of manufacturing method is strongly affected by material and heat treatment considerations. Forging, casting, and welding are limited by the types of material which can be used. Heat treatment tends to warp parts by relieving residual stresses or by introducing new ones through rapid quenching. Uniform cross sections reduce this tendency. Some components require a rough machining operation prior to heat treatment which is followed by a final machining operation after heat treatment. Material strength as well as material type also affect tool life and the speeds and feeds which can be employed in the metal cutting operations, thus, directly affecting the rate and cost of production.

2-5.6 DURABILITY CONSIDERATIONS [44-46]

Factors which affect the durability of components or assemblies include fatigue, wear, corrosion, and environmental effects. The manner in which each affects durability depends upon the application or service of the item, so that general durability considerations are difficult to delineate. The value engineer, however, must ascertain the effect upon the durability of the item, of any change in the part or in its environment (such as operating at different speeds or stress levels) either by conducting a thorough analysis or by using appropriate empirical data.

2-5.7 DESIGN BALANCE

The principal contribution of value engineering in obtaining a design balance lies in the systematic value analysis of the design. If designers could have sufficient time at their disposal to consider all of the design ramifications during the design phase, the value engineer's usefulness would be minimal. Since the designers do not have unlimited time available, however, they concern themselves more with the functional aspects of the design than with maximum value considerations. The inclusion of value engineers in the project group adds materially to the group's effectiveness by supplementing the designers with assistants trained in cost reduction through value analysis.

A design balance depends upon the proper planning and execution of analysis and design in the areas of performance, function, reliability, durability, maintainability, producibility, and cost. One area cannot be optimized at the expense of other areas without degrading the overall system design.

2-6 HUMAN FACTORS ENGINEERING

2-6.1 GENERAL CONSIDERATIONS

Human factors engineering probably had its start when the cave man discovered that he could use a club to extend his reach to obtain an advantage in hunting and combat. As man progressed, he discovered an ever-increasing number of devices with which to complement and supplement his natural resources, to enable him to live an easier life, or to give him an edge over his adversaries in combat. In a modern society, it is easy to take mechanization for granted and to forget that every machine, every piece of communication equipment, and every item of food and clothing is made for the benefit of man. As technology advances, man is required to perform more tasks, at a faster rate, and with more judgment than ever before. Therefore, as the human functions become more taxed the equipment designer requires a greater understanding of the human body in its machine environment in order to design optimum equipment.

Human factors engineering, then, considers the human body in relation to its mechanical and climatic environment. Consideration is given to such factors as body dimensions (anthropometry), the range and characteristics of bodily movements, the forces that the various body parts are capable of exerting, reaction times, fatigue, recuperation, boredom, anxiety, and physical comfort. A great deal of research

and study has been done in this field and much literature is available on this subject[49-56]. Refs. 49 and 51 are particularly useful to the design engineer as a source of design data and for a basic understanding of the incorporation of human capabilities into the system design. Ref. 50 is a compilation of general design practices for human factors engineering in the design of military vehicles. Ref. 52 is a directory of the Human Engineering Laboratories' publications which serves as a ready reference list for particular human factors engineering design problems and areas.

2-6.2 CREW COMFORT

2-6.2.1 Heating, Cooling, and Ventilating

The factors affecting the comfort of individuals—excluding air odors and cleanliness—are temperature, humidity, and the rate of air circulation. Since individuals vary in their sensations of comfort, there is no sharply defined comfort level; and it is, therefore, normally expressed as a range of several parameters based on statistical samplings of the population. Even population comfort levels tend to vary, depending upon their climatic locality; people from polar regions tend to be comfortable at lower mean temperatures than people from the equatorial regions. For personal safety and efficiency, however, all occupied spaces should be adequately ventilated--a minimum of 15 to 20 cfm is required.

In occupied or personnel space—the space of primary concern in the design of bodies and hulls—optimum temperature for personnel is somewhat a matter of personal preference and varies according to the nature of the tasks being performed, the clothing being worn, the relative humidity, and the air velocity. The quantitative environmental factors that relate to physical comfort are the dry- and wet-bulb temperatures of the air and its velocity relative to the subject. These three factors collectively in various combinations produce conditions of either comfort or discomfort. Discomfort may be the unsatisfactory sensation of either heat or cold.

An arbitrary index, termed "effective temperature", combines into a single value the degree of warmth or cold felt by the human body in response to the ambient air temperature, moisture content, and velocity. It is defined as the temperature of saturated still air (velocity of 15-25 ft/min due to natural turbulence) that induces the same sensations of warmth or coolness as those produced by the air surrounding the subject. Air velocity has a distinct bearing on effective temperature—increased velocities lower the effective temperature when all other conditions remain unchanged.

Fig. 2-1 shows the effective temperatures for combinations of dry- and wet-bulb temperatures and air velocities. By joining selected dry- and wet-bulb temperatures, located on their respective scales on Fig. 2-1, with a straight line such as line AB, the effective temperatures at various air velocities can be determined at the intersections of this line with the various air velocity curves. Similarly, the nomograph can be used to determine combinations of dry- and wet-bulb temperatures that will result in any desired effective temperature at selected air velocities. Optimum conditions for summer and winter are indicated on the figure. Maximum temperatures and corresponding maximum relative humidities for reliable human performance during prolonged exposure are shown in Table 2-1.

Prolonged exposure of the ungloved hand to temperatures below 55°F often results in a stiffening of the fingers, thus degrading performance in tasks requiring finger dexterity. Adequate clearance must be provided for a gloved or mittened hand, and due allowances must be made for the impaired tactile sense and dexterity caused by the hand covering.

Human comfort under cold conditions is greatly affected by the wind, in addition to the temperature, and is measured in terms of "windchill." Windchill is a measure of cold discomfort and is derived from the freezing rate of water when influenced by wind as well as low ambient temperatures[57]. It can be determined by means of the following equations or from Fig. 2-2:

$$K_a = (10\, W_v^{1/2} + 10.45 - W_v)(33 - T_a) \qquad (2\text{-}5)$$

where

K_a = windchill, kg-cal/m^2-hr

W_v = wind velocity, m/sec

T_a = ambient air temperature, °C

The same value can be obtained, using English units, from the following expression:

$$K_a = 0.248\ (14.95\ W^{1/2} + 23.38 - W)\ (91.4 - T) \quad (2\text{-}6)$$

where

W = wind velocity, mph
T = ambient air temperature, °F

Cooling values plotted in Fig. 2-2 are based upon heat flow in complete shade; for bright sunshine, allow for incoming solar radiation by decreasing values by 200 kg-cal/m²-hr, and for light clouds, decrease the indicated values by 1000 kg-cal/m²-hr.

There are many good references to aid the designer in the mechanics of designing an adequate heating and ventilating system. Some of this material is included in Refs. 58—63. Ref. 49 is also useful, since it discusses environmental effects on the human body.

TABLE 2-1 EXPOSURE (TIME) LIMITS FOR EXTREME ENVIRONMENTAL HIGH TEMPERATURES FOR MEN AT WORK IN OCCUPIED OR PERSONNEL SPACE

Environmental Temperature, °F	*Max Relative Humidity, %*	*Safe Exposure for Men at Work hr*	*Reference*
85	100	2	MIL-STD-803A-2 (USAF)
90	100	1	
95	100	0.5	
100	60	1.5	
105	60	1.0	
110	60	0.5	

2-6.2.2 Oxygen, Carbon Dioxide, and Carbon Monoxide[49]

Normal dry air contains 21 percent oxygen and 0.03 percent carbon dioxide. The minimum allowable oxygen content can be as low as 12 percent and the maximum allowable carbon dioxide content can be as high as 3.0 percent without incurring any long-term physical impairment; although a period of adaptation may be necessary. Engineering control of oxygen and carbon dioxide depends upon the size of the space to be ventilated and the number of occupants. The oxygen consumption of a resting adult male is 0.73 ft³/hr and his CO_2 production is about 0.61 ft³/hr. For an adult engaged in light physical activity (submariner), these figures become 0.90 and 0.74 ft³/hr, respectively. An ideal engineering control is one that maintains the CO_2 level below about 1 percent, which is well below the minimum level, and maintains the oxygen level well above the minimum allowable of 12 percent. The minimum ventilation of 15 to 20 cfm, previously recommended, will ensure a proper atmosphere.

Carbon monoxide (CO) is an odorless gas that is the product of incomplete combustion. While the tolerance dosage of CO varies from person to person, it is recommended that 0.003 percent CO be used as the standard permissible limit for long-term exposure.

2-6.2.3 Illumination and Visibility Requirements[49-51]

The area of illumination and visibility is an important part of the field of human engineering because without proper cognizance of the problem area associated with vision the effective utilization of the vehicle capabilities can be seriously limited. The general area of vision can be broken down into three interrelated areas—(1) exterior lighting, (2) interior illumination, and (3) visibility. The exterior lighting is discussed in Refs. 64 and 67. Ref. 64 contains information regarding the types of headlights, taillights, stoplights, reflectors, and turn signals required for normal night operation. Ref. 67 discusses the general characteristic

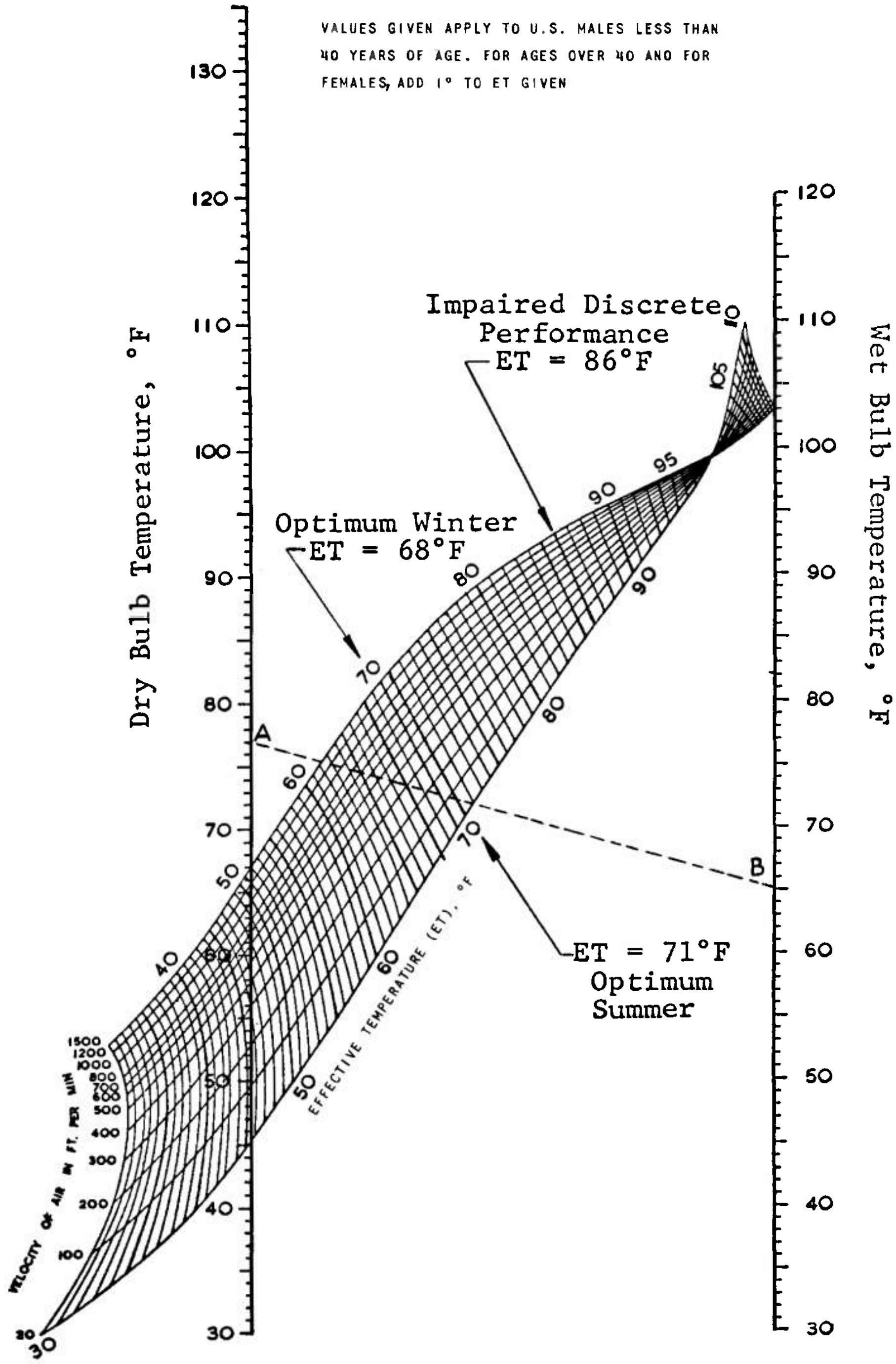

Figure 2-1. Effective Temperature Chart Showing Normal Scale of Effective Temperature, Applicable to Inhabitants of the U.S. Under the Following Conditions:
Clothing: Customary Indoor Clothing
Activity: Sedentary or Light Muscular Work
Heating Methods: Convection Type, i.e., Warm Air, Direct Steam or Hot Water Radiators, Plenum Systems

Reprinted by permission from *ASHRAE Guide and Data Book*[63].

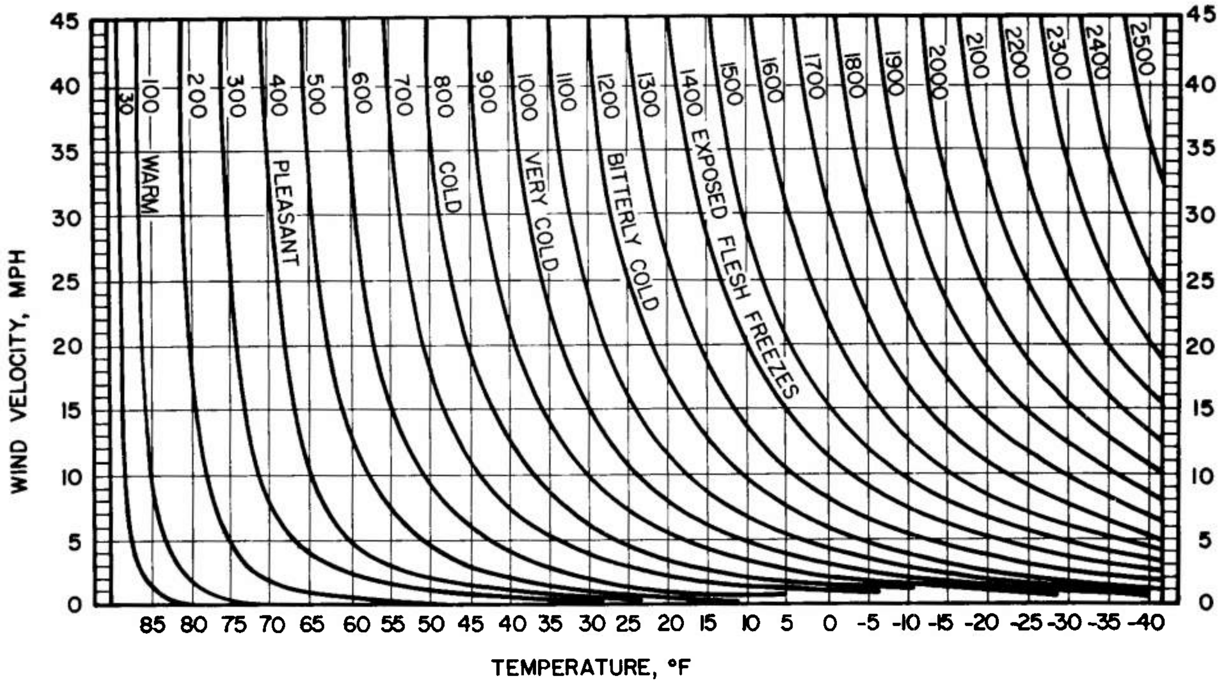

Figure 2-2. Windchill Factor Chart [57]

requirements of night performance for military vehicles.

The interior illumination designer must consider design areas such as interior colors, contrasts between instrument lighting and the surroundings, light intensities required to perform various tasks, blackout illumination requirements, glare, shadow, dial sizes, etc. In general, service lighting should provide illumination adequate and proper for the work to be done. General illumination should be at least 35 foot candles at desk or bench height (30 in.) with specialized lighting provided where required. The lighting system should have a dimming capability where various levels of illumination are required, e.g., in a shelter where radar operators monitor cathode-ray tube displays. Fluorescent lighting is preferred; however, this should be supplemented by sufficient incandescent lighting to permit "start of operation" at low temperature. Suitable blackout provisions should be provided for at all points of hull or shelter where light on the inside could become visible on the outside. Refs. 49-51, 67, and 68 offer some insight into the general area of illumination.

Visibility requirements of military vehicles are discussed in Refs. 50, 65, and 66. Reference 50 is primarily useful for the design of tactical vehicles, Ref. 65 covers both combat and tactical vehicles, and Ref. 66 is useful in the design of combat vehicles. Items considered include driver visibility requirements, mirror positions, windows, and door posts.

2-6.2.4 Seating Arrangements [50,51,53,54]

The design of work seating covers a wide scope, from the simplest of benches to the most elaborate of adjustable chairs. Proper seat design can reduce fatigue and promote increased productivity by its occupant; it can save time and energy. Poor seating may interfere with optimum operation of equipment and cut down the efficiency of the operator. The proper seat design should:

a. Provide a supporting framework for the body relative to the activity in which it is engaged.

b. Be conveninent to the task of the worker. The seat should be of proper size and should be

adjustable not only in height but in its position (forward or backward) as well.

c. Support the body properly to avoid poor posture. Cushions should be used to distribute body weight evenly over the surface of the seat.

d. Provide arm rests when they do not interfere with the individual task at hand; foot rests should be provided to maintain optimum seat-to-foot-rest distance.

Where high density seating is required, such as in personnel carriers, seating is made simpler by creating a continuous seat. This allows a small passenger to occupy only as much space as he needs while a larger passenger can benefit from the larger space thus made available. This type of seating, however, should be used only for short-haul transportation because it sacrifices some comfort.

General seating design information can be founds in Refs. 53 and 54. Ref. 50 contains general design data and miscellaneous seating considerations. Ref. 51 gives a more comprehensive discussion of seating design including a wide variety of seats suitable for military seating applications.

2-6.2.5 Doorway and Hatchway Requirements [49-51]

Properly designed and located doorways and hatches have a definite influence on the morale of crew and passengers. Their size should permit easy passage of the 99th percentile military man who is wearing bulky clothing and carrying—as a minimum—a pistol, canteen, and first aid pouch on his belt. In certain cases, provisions must be made for the easy access of men carrying loaded packs on their backs and carrying weapons or other equipment in their hands. Vehicles designed for use in the arctic should make adequate provision for the entrance and exit of personnel wearing bulky arctic clothing.

Doorways and hatches must be free from sharp edges and projections that might snag clothing or equipment. Door handles and latching mechanisms must be simple, foolproof, rugged in construction, and capable of being operated readily and quickly by heavily gloved or mittened hands. Hinges must be so constructed as to be free working under adverse conditions of mud, dust, sand, and ice. The hinges of armored vehicles must be either armored or so constructed as to be highly resistant to damage from enemy fire and have sufficient clearance between moving parts to prevent their immobilization by wedged projectile particles. Joints around hatches of armored vehicles must be designed to prevent bullet splash from entering the vehicle.

Escape hatches, being emergency exits, are not required to afford the same ease of exit as do the main doors and hatches; but they should be adequate to permit rapid exit without the necessity of removing any clothing or equipment normally worn inside the vehicle. In general, and to the extent possible and still meet other requirements, escape hatches should meet the requirements which follow:

a. Be conveniently located to the using personnel.

b. Be easy to open and with a minimum moving of equipment out of the escape hatch area.

c. Be located in areas of the vehicle that are least likely to be damaged by enemy fire.

d. Be located in areas of the vehicle that are least likely to come under enemy fire during the attempted escape. Escape routes should take maximum advantage of the vehicle structure for cover. Toward this end, the best locations for escape hatches are the bottom, the rear, and the sides of the vehicle. The top of the vehicle should generally be avoided for an escape hatch location (see exception in next paragraph) for several reasons: (1) it is often the location of the regular hatches and placing escape hatches in the same general area would be a pointless redundancy; (2) personnel attempting an escape through the top of a vehicle are exposed to enemy fire; and (3) when the escape is from a burning vehicle, the opening of a hatch in the top can produce a stack effect to intensify the fire and the personnel will be trapped by the flames.

Fully enclosed amphibious vehicles of the M113 type (Fig. 1-25) have normal passenger access through a large, downward opening door at the vehicle rear. This main access door is almost totally submerged during waterborne operations, and opening it to effect an emergency escape would be disasterous. This leaves the vehicle top as the only practical location for an escape hatch. However, consideration should also be given to escape from a capsized vehicle. Despite the low center of gravity and stability of amphibious vehicles, it

is possible for them to capsize due to improper loading, load shift, or especially when plunging into the water from a high bank. Since their hulls are in stable equilibrium when capsized (see par. 4-20), they will not right themselves. Escape hatches should be provided, therefore, that will permit escape under these conditions.

2-6.3 CREW PROTECTION

2-6.3.1 Ballistic Protection[69]

In general, the degree of ballistic protection considered desirable for combat vehicles varies inversely with the distance behind the forward edge of the battle area (FEBA) that its missions are expected to take it. Exceptions to this are vehicles that rely on their speed and maneuverability to reduce their vulnerability. When operating close to the FEBA, the vehicle is within range of the enemy's direct fire weapons; the probability of being hit is relatively high; and heavy armor protection is required against these weapons. In rear areas, the probability of attack by direct fire weapons is greatly decreased and the threat is from predicted fire weapons, indirect fire, and from attack by aircraft using small caliber high rate-of-fire weapons, cannon, rockets, or bombs. Although a direct hit from these weapons can be lethal to a vehicle, the probability of a direct hit is relatively low because of the difficulties encountered in these types of attack. This is the basis for reducing the amount of armor required on rear vehicles.

Unfortunately, there are no firmly established rules governing the amount of ballistic protection required by vehicles. A rule-of-thumb often applied to armored vehicles states they should have sufficient ballistic protection on their fronts and turrets to defeat a direct hit from a weapon equal to the ones with which they are equipped. The price of heavy ballistic protection, however, is high in terms of vehicle weight, size, power requirements, speed, maneuverability, as well as monetary costs. In order to minimize these, a degree of risk must be accepted. Following are some of the factors to be considered in this analysis:

a. The primary and secondary missions of the vehicle and their relationship to the FEBA.

b. The types and effects of the weapons that may be used against the vehicle.

c. The probability of a hit from various weapons and the specific areas of the vehicle that may be hit. Combat information has shown that certain areas of a vehicle are hit more frequently than other areas, and this varies with the type of weapons used. Thus, each area of the vehicle has a double probability of being hit; first the probability of being hit if a particular weapon is fired, and second, the probability of a particular weapon being fired.

d. The vulnerability of each particular area to a particular weapon. Merely sustaining a hit upon a particular area of the vehicle by a particular weapon may not by itself be decisive. Significant damage must be effected upon the vehicle or its crew by penetration, blast, shock, high-velocity fragments, or some combination of these to immobilize, destroy, or otherwise impair or neutralize the functional effectiveness of the vehicle. The degree to which weapon damage alters the effectiveness of the vehicle in accomplishing its mission is a measure of vulnerability. Since all parts of a vehicle are not equally sensitive to weapon damage, it is not necessary to provide all areas with equal protection.

AMCP 706-170[69] is extremely useful in the design of ballistic protection for vehicles. It describes the mechanics of ballistic damage, illustrates a method of obtaining balanced ballistic protection, and discusses ways and materials used in the design of ballistic protection.

2-6.3.2 Environmental Protection[114]

2-6.3.2.1 Sand and Dust[68]

Excess sand and dust reduce personnel efficiency and can cause physical harm if enough is accumulated in the respiratory tract. In open vehicles, personnel are usually protected by the use of goggles and dust masks. Protection of personnel through vehicle design is limited to normally closed vehicles. If the vehicle is equipped to withstand CBR attacks by using a filtered ventilation system, no additional sand and dust protective equipment is required. Otherwise, filters can be added to the ventilating system to trap the sand and dust. Since sand and dust particles are relatively large, the air filter can be made of an open cellular foam plastic material. This type of filter material is inexpensive, readily available, and can be re-used

by simply washing out the accumulated sand and dust particles.

2-6.3.2.2 Rain, Mud, Sleet, and Snow [68]

In order to maintain a desirable level of system efficiency, vehicle crews must be given a certain degree of protection against inclement weather. While it may be impractical to eliminate all inclement weather effects, vehicle design should minimize them. Thus crew protection against rain, mud, sleet, and snow must be considered in the design of splash guards, fenders, cabs and troop compartment enclosures, sump pumps for hulled vehicles, windshield wipers and defrosters, space heaters and air conditioners, and door and hatch locations.

2-6.3.2.3 Travel Hazards

With only a few exceptions, military vehicles have as either their primary or secondary missions the transporting of personnel. These personnel can be the vehicle's operating crew, its passengers, or both. Since human cargo is more sensitive to its environment than is inanimate cargo, the body designer should give due consideration to the special requirements of human cargo. Principal among these are the requirements imposed by road shocks, longitudinal and lateral accelerations, and extreme pitch and roll angles of the vehicle. These conditions can result in physical injury to personnel by causing them to strike objects within the vehicle, to become unseated and possibly even to be thrown from the vehicle, or to suffer injury due to the shock forces experienced by the body. Efficient seat belts, where practical, will keep personnel in their seats. Where not practical, as for troops riding on removable bench seats in a cargo truck, a resilient seat with an adequate backrest should be provided along with a comfortably placed footrest (this can be the floor) and some form of grab handles. The riding quality of automotive vehicles is primarily in the domain of the suspension designer; but since he, too, must make trade-offs and compromises, the body designer can assist by providing additional elastic support and adequate personnel restraint.

Excessive vibration is another consideration of the body designer. Vibrations experienced by the personnel within a vehicle result from the interactions of the vehicle's suspension system with the ground and the vibrations produced by various vehicle-mounted mechanisms such as engines (main and auxiliary), transmissions, final drives, pumps, compressors, generators, etc. These affect personnel by producing uncomfortable noise levels, general discomfort, increased fatigue, and a general decrease in operating efficiency. The body designer can usually reduce these undesirable vibrations to an acceptable level through the application of vibration isolators, stiffening techniques, vibration damping techniques, and sound deadening materials.

2-6.3.3 Chemical, Biological, and Radiological Agents

2-6.3.3.1 Chemical Agents [72,75]

The military gases and screening smokes have a corrosive effect upon metals; but in the concentrations generally used in military operations and the relatively short time that vehicles are exposed, the effect upon military vehicles is negligible, even when liquid sprays contact the metal. Perhaps a greater source of corrosive damage to vehicles is the various materials used in their decontamination after exposure to highly persistent chemical agents. Decontaminating agents may be strong caustic solutions, acids, oxidizers, or strong detergents, depending upon the chemical agent for which they are intended. Where a choice of structural materials is possible, preference should be given to those more resistant to the effects of decontaminants. Body and hull structures should be designed to facilitate decontamination. Smooth, downward sloping surfaces are more easily washed down than rough and flat horizontal surfaces. Blind pockets that are difficult to clean or that tend to hold solutions should be avoided.

Military gases are a threat primarily to personnel. Personnel riding in closed vehicles are afforded considerable protection by the vehicles—particularly from sprayed agents. Since most chemical agents are particulates, either liquid or solids, they can be kept out of the vehicle by installing efficient filters on the intakes of the ventilating system and sealing all

hatches, ports, and access openings. This can be developed into a completely self-sufficient life support system by incorporating equipment to remove carbon dioxide and carbon monoxide from the air, maintaining an adequate oxygen supply, and controlling the temperature within acceptable limits by heating or cooling. Filtration systems should be capable of removing particles of 5-micron diameter or less. Absolute filters are the most desirable.

The oxygen requirements can be met by filtering outside air or from high pressure oxygen bottles carried on board the vehicle. Several systems are available for the removal of CO_2. The most efficient of these is based upon the absorption of the gas by Baralyme. Baralyme is a crystalline caustic mixture with the following by weight composition: 20 percent (by wt) $Ba(OH)_2 \cdot 8H_2O$, 80 percent $Ca(OH)_2$, small amount of KOH, trace of Mimosa Z dye plus Ethyl Violet, and a trace of wetting agent. Theoretically, 2 lb of Baralyme will absorb 1 lb of CO_2, but experience has shown that from 3 to 5 lb are actually needed[73].

Careful consideration must be given to the seals used at the various hatches, ports, and other openings leading to the outside. It is practically impossible to achieve and maintain a perfectly sealed crew or passenger compartment when gases are concerned and where provisions for access and vision are required. Perfect fits are costly; sealing surfaces deteriorate due to wear and environmental factors; and gases can diffuse through many sealing materials. Faced with these facts, the designer might do well to pressurize the interior of the vehicle to a few psi above atmospheric pressure. In this way he will assure that leakage past the seals will be outward rather than inward most of the time.

During short periods of overpressure produced by high explosive blasts, contaiminated outside air may still be forced into the vehicle. For protection under these circumstances, the personnel will still need to wear gas masks and protective clothing. The pressurized interior can be purged of contaminated air by means of exhaust fans. Persistent agents (liquids or solids) that may be forced into the vehicle will have to be washed down with decontaminating agents. Thus, the interior surfaces should be designed to facilitate this cleanup.

2-6.3.3.2 Biological Agents[74,75]

Biological warfare is the tactical employment of living organisms, viruses, certain classifications of micro-organisms, toxic biological products, and plant growth regulators to produce death or casualties in man, animals, or plants and also the defense against such action.

Modern concepts of this type of warfare employ highly concentrated aerosols of infectious materials that can be disseminated in much the same manner as are chemical agents. Defense against this type of warfare consists of immunization of personnel, early detection of biological agents, the wearing of gas masks and protective clothing, decontamination of ground and equipment, and miscellaneous measures to avoid exposure to the agents.

Since the onset time for most diseases is a matter of days or weeks, biological agents will not be used deliberately against personnel within vehicles. However, vehicles will, in all probability, encounter biological clouds or contaminated terrain. Under these conditions, they can afford considerable protection to the personnel within if the general principles discussed under chemical agents (par. 2-6.3.3.1) are applied. The vehicle itself will require decontamination.

Decontamination from biological agents involves various sterilization techniques designed to kill the living organisms. The materials used for this purpose are formaldehyde, peracetic acid, sulfur dioxide, hypochlorite, and others. While the biological agents themselves have little effect upon vehicles, the decontamination agents cause corrosion of metal parts, etching and general deterioration of plastics, and a degradation of fabrics and elastomers. Vehicle bodies and hulls should be designed to facilitate the washing away of the decontaminating agents as quickly and as thoroughly as possible.

The air drawn into the vehicle's ventilating system must also be decontaminated when operating in contaminated areas. This can be done by passing it through absolute filters or by sterilizing it as it is drawn in. The filter elements require periodic sterilization in addition to normal cleaning, and the filtered particles must be thoroughly sterilized before they are

discarded. Sterilization can be accomplished with various chemicals[75], with steam, or by incineration—incineration is the most effective. When incineration techniques are applied to the ventilating air supply, the air requires cooling before it can be used. This places additional requirements upon the mechanical system of the vehicle.

2-6.3.3.3 Radiological Agents[75,76,78,79]

Radiological warfare is defined[77] as "the employment of agents or weapons to produce residual radioactive contamination as distinguished from the initial effects of a nuclear explosion (blast, thermal, and initial nuclear radiation)". Residual radiation is principally the radiation emitted by fallout deposited on the ground and equipment as a result of a nuclear detonation. To a lesser extent, it is also radiation from induced contamination, such as radioactive earth made radioactive by the neutron bombardment, and from fallout particles remaining airborne. Fallout may be deposited over very large land areas including locations where no other effects from the nuclear detonation have been experienced. Induced radioactivity, however, will be found only close to ground zero of the detonation.

Residual contamination consits of alpha and beta particles, and gamma radiation; neutron particles are only found during the initial radiation phase (the first minute after the nuclear burst). Of these three, only gamma radiation is of concern to the designer intent on providing shielding against residual radiation, because both alpha and beta particles can be stopped by very thin layers of suitable material.

Most residual radiation received by personnel within a vehicle is transmitted directly through the vehicle walls and floor, but a significant amount does arrive through the roof. This latter radiation is due to a scattering in the air caused by dust and moisture particles and referred to as "skyshine". Combat vehicles, particularly tanks and armored personnel carriers, inherently provide a considerable amount of shielding against residual radiation due to their armor and massive metal parts. The amount of shielding against residual radiation is almost a direct function of the mass of material between the individual and the radiation source. The effectiveness of material in attenuating radiation is represented by its "half-value thickness". This is defined as the thickness of a particular material required to absorb one-half of the gamma radiation falling upon it. Approximate half-value thicknesses for steel and aluminum are 0.7 and 1.9 in., respectively.

The amount of shielding provided for each crew member is expressed in terms of a "protection factor". This is defined as the ratio of the radiation level existing 3 ft above the ground in the open fallout field to the radiation level at the crew members' location inside the vehicle. Some heavy tanks have protection factors against residual radiation as high as 20 whereas a 1/4-ton truck has a protection factor of about 1.25 (Ref. 78). The term "transmission factor" is also used to express the amount of radiation shielding. This is esentially the inverse of the protection factor, or the ratio of dose rate inside the vehicle to the dose rate outside. Fig. 2-3 illustrates the transmission factor as a function of material and material thickness. The dosage received is equal to the dose rate outside multiplied by the transmission factor. Some typical transmission factors for military vehicles exposed to residual radiation are:[79]

Vehicle	Transmission Factor
Armored Personnel Carrier	0.6
Tank, light	0.1
Tank, medium	0.2
Truck, ¼-ton	0.1
Truck, ¾-ton	0.8
Truck, 2½-ton	0.7
Truck, 4- to 7-ton	0.5

Another radiation hazard to personnel inside a vehicle in a radioactive area is from radioactive fallout particles or particles of activated soil that may enter the vehicle. The principal danger here, in addition to the ability of these particles to raise the radiation level within the vehicle, is the possiblity of (1) these particles being inhaled or ingested by the crew, causing a serious internal hazard, and (2) the particles coming in contact with the skin and causing "beta burns" from beta radiation. Radioactive particles can enter the crew compartments of vehicles through access and vision openings, through clearances around weapons, turrets, cables, piping, and control rods, and through the ventilating system. This danger can be minimized by providing sealed closures for all openings, reducing all

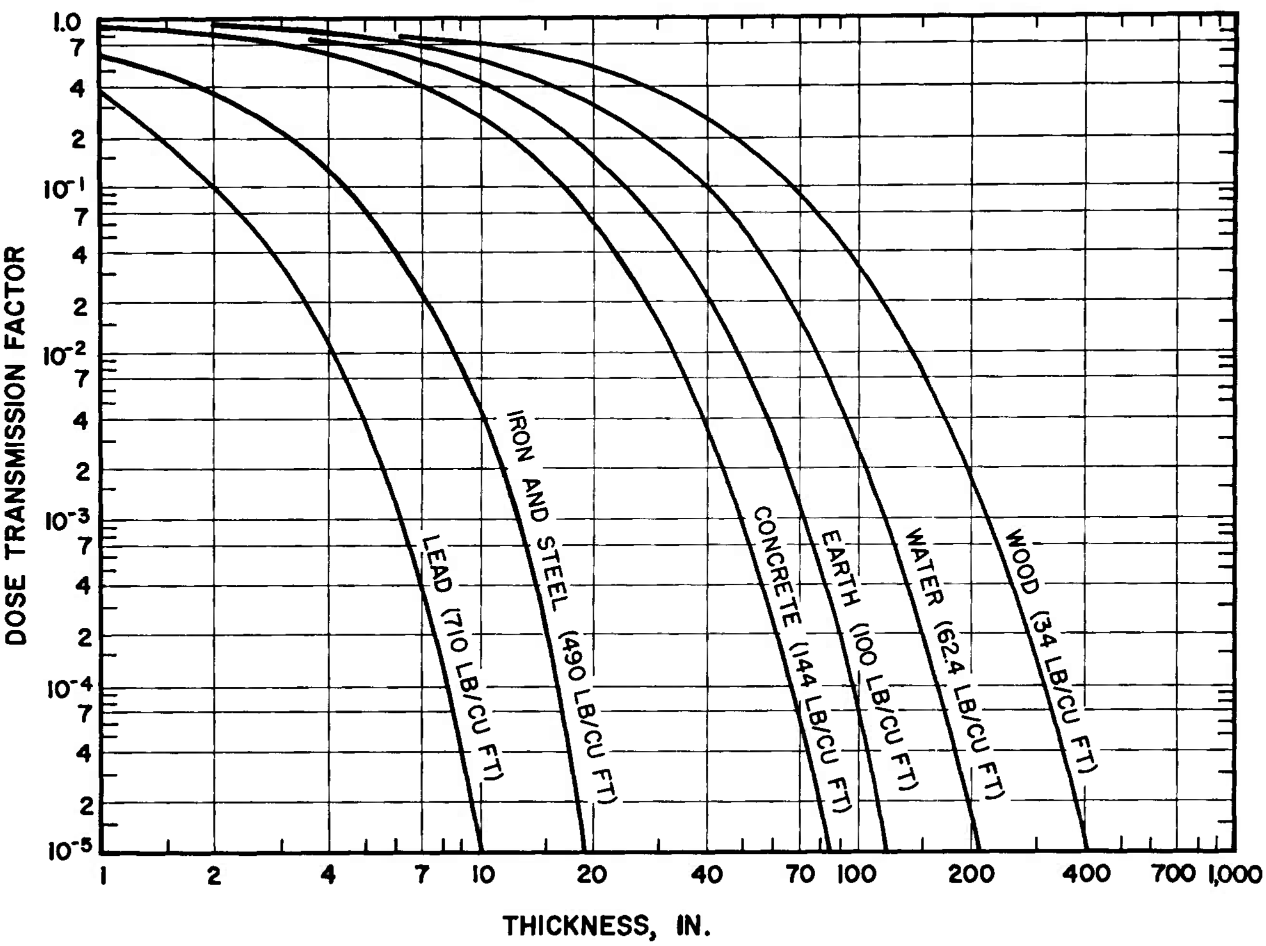

Figure 2-3. Dose Transmission Factors for Gamma Radiations of Various Materials[76]

clearances to a minimum and providing them with seals, equipping the ventilating system with adequate filters, and providing a positive pressure inside the vehicle.

In addition to residual radiation, previously considered, one must consider the damage mechanisms resulting from the initial phase of a nuclear weapon detonation when exposed materiel is subjected to blast, initial nuclear radiation, and thermal radiation. The blast effects create no new problems not experienced with non-nuclear weapons—the difference is in the magnitude of the blast. Initial nuclear radiation consists of alpha, beta, and neutron particles; and gamma rays.Only the gamma rays and neutron particles are of concern during the initial phase because of the very short ranges of the alpha and beta components. Shielding against initial radiation, particularly at relatively close ranges, is extremely difficult.

While most Army equipment is designed to survive the conditions that exist on the conventional battlefield, only moderate consideration is given to the problem of survival on the nuclear battlefield. This attitude is largely due to the more pressing requirements of the conventional battlefield and to a reluctance to burden the materiel with the extra weight and expenditure of funds necessarv to assure a reasonable degree of nuclear battlefield survivability—especially since the crew would, in most cases, be more vulnerable than the materiel. Nevertheless, a certain degree of invulnerability to the effects of nuclear weapons is expected of most materiel. Guidance on the degree of invulnerability required is normally given in the pertinent Qualitative Materiel Requirements (QMR). When this is not available, Refs. 80 and 81 may be used as a guide.

Tank-type combat vehicles, being more

sturdily constructed than other vehicles, can survive the blast and thermal effects of a nuclear weapon at relatively close ranges. At these ranges, they can experience neutron flux densities sufficiently intense to produce induced radiation in the structural materials of the vehicle. Under these conditions, some of the materials so affected will emit high-energy gamma radiation for a significant time after the neutron pulse to be a serious hazard to personnel that may later occupy the vehicle or to maintenance crews.

Sodium and silicon, two constituents of glass, form the radioactive isotopes sodium-24 and silicon-31 which have half-lives of 15 and 2.6 hours, respectively. They require 115 and 18 hours, respectively, after the irradiating pulse for their activity to decay below one percent of the original values. Manganese, a common constituent of many steels, forms the isotope manganese-56 which also has a half-life of 2.6 hours. Because of the preponderance of steel over sodium-bearing glass in a combat tank, the manganese in the steel may constitute a more serious hazard than sodium during the first few hours after the explosion despite its much shorter half-life.

Other metals that may become sources of induced radioactivity are zinc, copper, and, to a lesser extent, iron. Again, consideration must be give to the preponderance of iron in a tank. Aluminum can form the radioactive isotope aluminum-28, but its half-life is only 2.3 minutes. Thus, a vehicle containing a considerable amount of aluminum, as in an aluminum hull, could become highly radioactive initially, but very little would remain after one hour of the explosion. Some elements—e. g., boron—absorb neutrons without becoming radioactive, and their presence decreases induced radioactivity.

Shielding against neutrons presents a more difficult problem than shielding against gamma radiation. Where high-density materials are effective against gamma rays they are not as effective against neutrons when used alone. The attenuation of neutrons from a nuclear explosion involves several different phenomena. First, the very fast neutrons must be slowed down into the moderately fast range; this requires a suitable (inelastic) scattering material such as the one containing barium or iron. Then, the moderately fast neutrons have to be decelerated (by elastic scattering) into the slow range by means of an element of low atomic weight. Water (or other hydrogenous materials) is very satisfactory in this respect since its two constituent elements, hydrogen and oxygen, both have low atomic weights. The slow neutrons must then be absorbed. This is not a difficult matter since the hydrogen in water will serve the purpose. Unfortunately, however, most neutron capture actions are accomplished by the emission of gamma rays. Consequently, sufficient gamma-attenuating material must be included to minimize the escape of the gamma rays from the neutron shield. For additional discussions on neutron shielding techniques consult the pertinent references given at the end of this chapter, e. g., Ref. 115.

Thermal radiation, the third damage producing factor which occurs during the initial phase of a nuclear detonation, can be injurious to personnel at distances where blast and nuclear radiation have little effect. It is desirable, therefore, that vehicles provide some protection from this radiation.

Thermal radiation consists of ultraviolet, visible, and infrared radiations originating in the fireball. Thermal radiation travels with the speed of light, but has very little penetrating power and can be easily absorbed or reflected. Any solid, opaque material—e.g., a vehicle wall or a tarpaulin—between the fireball and a person or object will provide protection from this radiation. Conversely, transparent materials, such as glass, provide almost no protection. Tarpaulins or other external members used for this purpose should be made of nonflammable materials or should be treated to make them nonflammable.

Flash blindness is another matter of concern. The flash of light from a nuclear explosion can dazzle personnel, particularly at night, causing temporary loss of vision or even permanent retinal burns. These effects can be produced at greater distances than skin burns. Eye protection is expected to be afforded personnel who use the vision devices on radiologically protected vehicles. These devices, in response to the burst, instantaneously create an opaque barrier in the viewing system. Less effective means of protection may be obtained by vision devices that limit the field of view thereby decreasing the probability of seeing a nuclear flash.

2-7 CARGO ENGINEERING

2-7.1 GENERAL CONSIDERATIONS

Military cargo is generally transported in the theater of operations on tactical vehicles of various types described in Chapter 1. Typical military cargos—in addition to weapons, ammunition of all types, and personnel— include such commodities as rations, fuels, lubricants, water, liquid gases, electronic equipment, instruments, machinery, construction materials, bridge equipment, quartermaster supplies, and a multitude of other items. These may be transported loose or packed in boxes, crates, cans, or barrels. Normally, the items are packaged in some type of container to facilitate handling, however, there are no standard containers for all items. Similar items are generally packaged in similar containers so that, for example, all 105 mm howitzer projectiles are boxed in the same type and size container.

In a broad sense, every vehicle carries some type of cargo; but in the sense used here, cargo carriers are a family of wheeled and tracked vehicles designed for the primary mission of transporting bulk types of cargo. The various types of cargo carriers are discussed in considerable detail in Chapter 1.

2-7.2 SPACE AVAILABILITY AND STOWAGE[82]

On 25 Oct. 1960, Headquarters U. S. Continental Army Command published the results of a comprehensive study of cargo vehicle requirements of the field armies entitled, *Mover; Motor Vehicle Requirements, Army in the Field, 1965-1970.* This study recommends the development of a fleet of tactical vehicles that is expected to satisfy all anticipated Army needs. The payload capacities and cargo bed dimensions of the recommended vehicles are given in Table 2-2.

TABLE 2-2 CHARACTERISTICS OF VEHICLES RECOMMENDED BY MOVER STUDY

Vehicle Type	*Payload, Tons*	*Size of Cargo Bed, ft*	
		Regular	*Long*
Trucks	1/4	5 X 5	None
	1 1/4	8 X 7	12 X 7
	2 1/2	12 X 8	None
	5	14 X 8	None
	8	10 X 9	18 X 8
	16	15 X 9	18 X 8
Trailers	1/4	6 X 5	None
	1/2	9 X 6	None
	1 1/4	9 X 6	None
	1 3/4	Tank {500 gal fuel / 400 gal water	None
	2 1/2	9 X 6	None
	5	15 X 8	None
	10	23 X 8	None
Semitrailers	3 1/2	20 X 8	None
	7 1/2	30 X 10	None
	12	31 X 10	49 X 10
	25	30 X 10	None
	60	34 X 10	None

2-7.3 LOADING AND UNLOADING

Present standard military vehicle cargo deck heights vary from from 33 in. for the M37, 3/4-ton, 4 X 4 Cargo Truck to 56-7/8 in. for the M55, 5-ton, 6 X 6 Cargo Truck. The height of the cargo deck is not, as yet, standardized and varies from vehicle to vehicle depending upon the size, the design, and the road clearance requirements. Since the energy required to load or unload cargo is directly proportional to the height the cargo must be lifted, the height of the cargo deck should be made as low as possible, consistent with the other vehicle requirements, to facilitate loading and unloading.

In order to maximize the loading and unloading efficiency, the tailgate of the vehicle should be capable of being dropped or swung out of the way in such a manner as to leave the cargo deck unobstructed. Vehicles equipped with sponsons or extended wheel wells may require a second tailgate to fully clear the cargo deck. In addition, all troop seats should be capable of being stowed or folded so as to leave a cargo compartment free from obstructions. If the cargo deck is abnormally high, hand grips and steps or foot holes should be provided to aid the cargo loaders in getting the cargo up onto the cargo deck.

2-7.4 CARGO PROTECTION

Cargo is packaged[83] in such a manner as to be protected from normal handling loads and environmental effects. The vehicle should be designed in such a manner that the normal capabilities of the package are not exceeded when the vehicle is properly loaded. The body or hull must provide the following:

a. Protection of the cargo from inclement

weather. While the cargo packaging is designed to resist the effects of inclement weather, the amount of cargo exposure should be minimized. This type of protection is normally afforded by providing a removable waterproof roof which extends over the top, sides, front, and back of the vehicle, or by totally enclosing the vehicle cargo compartment with Fiberglas or metal. The removable roof should project over the sides of the vehicle so that normal water run-off does not leak into the cargo compartment.

b. Protection of the cargo from more than the normal amount of road dust, mud splash, and water. Cargo protection from abnormal road dust is generally accomplished by placing an extra tarpaulin over the cargo. Mud protection is provided by properly designed fenders, mud guards, and sponsons. Water protection is provided by adequate freeboard, for amphibious vehicles, and by properly designed cargo compartment seals.

c. A minimum of load concentrations acting on the cargo. This is provided by flat floors and sides, and by designing cargo areas with a minimum of irregular projections and obstructions. Floor runners and skid strips may be used on cargo floors; however, these should be made as low as possible and spaced so that packages can either be placed between them or be supported by at least two runners.

d. In some cases, shade, ventilation, heating, cooling, or refrigeration may be required to keep cargo from deteriorating. The solar radiation can raise the local package temperature far above the ambient temperature and exceed the temperature protection capabilities of the package unless open shade or ventilation is provided. If the vehicle has a removable cargo closure, the sides or ends can be removed to provide some degree of ventilation. Solidly enclosed cargo vehicles normally have mechanical ventilating systems which are adequate while operating. For arctic conditions, temperature-sensitive cargo can be protected against freezing by the addition of space heaters placed in the cargo compartment. These heaters may either utilize the vehicle fuel supply or operate from separate fuel sources.

2-8 MATERIAL SELECTION

The selection of a material for use in military vehicles is largely dependent upon the understanding of the environment in which the vehicle or part is to be used and the properties of the candidate materials available for use in the particular application. The environment cannot be overemphasized since without a substantial knowledge of the loads, vibrations, shocks, temperatures, and other environmental parameters, the part cannot be properly designed. The selection of the material is equally important since the best paper design is worthless without the proper material from which to make the part.

As material technology advances, the selection of candidate materials for any particular application offers an ever-widening choice of materials and fabrication techniques. Lightweight metals-such as aluminum, titanium, and magnesium—are competing with, and surpassing, steels for many material applications. Plastics offer a cost, weight, and functional advantage over metals in some applications. Material selection, thereore, requires not only a knowledge of what material is available to the designer, but the designer must also have an understanding of material properties such as durability, cost of material, cost of manufacture, strength, weight toughness, resilience, ductility, malleability, and hardness if a proper marriage between the material and its environment is to be achieved.

2-8.1 METALS

2-8.1.1 Strength[44]

Strength is the ability of a material to resist loads without yielding or fracturing. A metal may have good elastic strength, good plastic strength, or both. *Elastic strength* is a measure of the ability of the metal to deform under load and return to its initial state when the load is removed. *Plastic strength* is a measure of the ability of a metal to deform plastically without fracture. Elastic strength is the yield point stress (sometimes the proportional limit stress) and its units are psi. Plastic strength is the ultimate stress of the material and its units are also psi.

2-8.1.2 Stiffness[44]

The mechanical property of a material to resist deformation in the elastic range is called stiffness. One measure of stiffness is the modulus of elasticity E and its units are psi. This

modulus may be defined as the material unit stress (lb/in.^{2}) divided by the material unit strain (in./in.), in the linear elastic region of the stress-strain curve. A nonlinear relation normally uses a 0.002 or a 0.0035 strain offset to define elastic modulus. It may be recalled from strength of materials studies that the elastic deflection of a body, such as a beam, is a function of the beam shape, the load, $1/E$; so that, for two beams having the same shape and load, the beam made of the material with the largest modulus of elasticity will have the smallest deflection.

2-8.1.3 Ductility and Malleability[84]

The term *ductility* denotes the capacity of a material for plastic deformation in tension or shear without fracture. *Malleability* denotes the capacity of a material for plastic deformation in compression without fracture. Ductility is usually expressed in terms of percent elongation at fracture. No general method applicable to all metals has been devised for measuring malleability, although this property is of importance in rolling, forging, and similar operations.

2-8.1.4 Resilience[44,84]

Resilience is the capacity of a material to absorb energy in the elastic range. A measure of resilience is the *modulus of resilience* U_r which is defined as the maximum amount of energy that a unit volume of material will store under load and release completely when the load is removed. It can be expressed mathematically as:

$$U_r = \frac{S_p^2}{2E} \tag{2-7}$$

where S_p is the proportional limit stress (psi) and E is the modulus of elasticity. Mild steel exhibits a U_r in tension of approximately 17 in./lb/in.^{2}. The total energy that a member can absorb without exceeding the proportional limit is the product of the modulus of resilience and the volume of the material being stressed. In some cases the proportional limit stress cannot be accurately determined, and is approximated by using the yield-point stress in place of the proportional limit stress.

2-8.1.5 Toughness[44,84]

Toughness is the capacity of a material to resist fracture under dynamic loading. Toughness is sometimes measured by the *modulus of toughness* U_t which is the amount of strain energy per unit volume absorbed up to rupture.

$$U_t = \int_0^{\epsilon_r} S\, d\epsilon \tag{2-8}$$

where S is the stress, ϵ is the strain, and 0 to ϵ_r are the integral limits from no strain to the strain at rupture. U_t, therefore, represents the total area under the stress-strain curve to the point of rupture. An approximate, but more convenient, measure of toughness for ductile materials is called the *toughness index number* or *merit number* T_o. $T_o = S_u \epsilon_r$ where S_u is the ultimate stress at rupture and ϵ_r is the ultimate strain at rupture.

2-8.1.6 Hardness[84,85]

The term hardness is used to designate several qualities. Hardness may indicate resistance to abrasion, cutting, or shaping; it may denote strength, stiffness, brittleness, resilience, or toughness; or it may denote combinations of these qualities. Its meaning is dependent upon the material used and the environment of the material. The vagueness of the definition is reflected by the multiplicity of hardness measuring instruments in general use. Some common hardness indices are the following:

a. *Brinell number.* In the Brinell test, a standard load is applied to a flat surface of the metal being tested for 15 sec by means of a steel ball 10 mm in diameter. The Brinell hardness number is evaluated as the load applied divided by the area of the spherical indentation remaining after the load is removed.

b. *Rockwell number.* In the Rockwell test, a 1/16-in.-diameter hardened steel ball (Rockwell B) or a standard diamond point (Rockwells A and C) is pressed into the material by a standard load. The depth of penetration is used as an indicator of the Rockwell number.

c. *Knoop hardness number.* The Knoop hardness number is also evaluated by an indentation test. The Knoop indentor is a diamond with a pyramidal point to produce a diamond-shaped indentation with a ratio of diagonals of 7.11 to 1. After the load is removed, the length of the long diagonal is measured, and the Knoop hardness number is computed as the ratio of the load to the projected area of the indentation.

d. *Other methods.* Other hardness measurement methods include the Vickers method, the Scleroscope method, the Monotron method, the Herbert pendulum method, the Herbert cloudburst tests, and a variety of mutual indentation methods. Details of these methods can be found in Refs. 84 and 85.

2-8.1.7 Commonly Used Metals

2-8.1.7.1 Steel[86-88,116]

Steel can be defined broadly as the material resulting from a process of removing impurities from scrap and pig irons and adding certain elements in predetermined amounts to impart desired properties to the resulting metal. Steels are arbitrarily divided into two categories; namely, carbon steels and alloy steels.

Carbon steels are steels whose physical properties are derived primarily from the percentages of carbon they contain and in which the quantities of other alloying elements are considered to be negligible. All carbon steels contain small quantities of certain residual elements, such as copper, nickel, molybdenum, chromium, etc., that were unavoidably retained from the raw materials. These are neither specified nor required and are normally considered as incidental.

The American Iron and Steel Institute (AISI) defines carbon steels as steels for which no minimum content is specified nor required for the alloying constituents of aluminum, boron, chromium, cobalt, columbium, molybdenum nickel, titanium, tungsten, vanadium, zirconium, or any other element added to obtain a desired alloying effect; for which the specified minumum copper content is not more than 0.40 percent; and for which the specified maximum content of any of the following elements is not more than the percentages noted: manganese, 1.65; silicon, 0.60; and copper, 0.60.

Alloy steels are steels to which one or more alloying elements have been deliberately added in significant quantities with the object of conferring particular properties to the steel. The AISI defines alloy steels as steels for which either (1) the maximum of the range given for the content of alloying elements exceeds any of the following percentages: manganese, 1.65; silicon, 0.60; copper, 0.60; or (2) for which a definite range or a definite minimum quantity of any of the following elements is specified or required within the limits of the recognized field of alloy steels: aluminum, boron, chromium (up to 3.99 percent), cobalt, columbium, molybdenum, nickel, titanium, tungsten, vanadium, zirconium, or any other alloying element that is added to obtain a desired alloying effect.

In general, alloy steels are more difficult to produce than are carbon steels, because their chemical contents are more closely controlled and special soaking-pit practices and careful slow-cooling in the billet form are required to prevent flakes or cracks. Nickel steels have good toughness, economical heat-treating procedures, and good corrosion resistance. They are especially suitable for case hardening. Chromium steels provide depth hardening capabilities and good wear resistance. Molybdenum steels provide good hardenability control, corrosion resistance, and are less susceptible to temper brittleness than alloy steels without molybdenum. Vanadium steels have a refined grain structure and good mechanical property control.

Steels are usually chosen for their high strength, good toughness, fatigue resistance, corrosion resistance, formability, and weldability. In many design problems, steel is the only material which is suitable; in others, where minimum weight is most important, other metals, such as aluminum, are used to advantage.

2-8.1.7.2 Aluminum[89,91]

Aluminum owes most of its applications to its light weight, relatively high strength, good corrosion resistance, good working properties, and good electrical and thermal conductivities. Aluminum is not generally used in its pure form but is alloyed with other metals. It may contain as much as 20 percent of other metals such as copper, magnesium, silicon, and zinc. The specific alloy used depends upon the physical properties desired in the metal. Through alloying and heat treating aluminum strengths in the order of five times that of the pure metal can be achieved.

The common or nonheat-treatable aluminum alloys contain constituents that remain substantially in solid solution or are insoluble at all temperatures. This group contains the

high-purity aluminum and wrought alloys of manganese, and magnesium (1000, 3000, and 5000 series). The strength of these alloys is increased by cold working. The strong or heat-treatable alloys contain alloying constitutents that have increased solid solubility at elevated temperatures. These are alloys of copper, magnesium silicon, and zinc (2000, 6000, and 7000 series). The strength of these alloys is increased by solution heat treatment.

The excellent adaptability of aluminum alloys to all forming methods has been a major factor in its adaptation by industry. High purity aluminum has the best formability, but the high-strength alloys are quite readily formed before heat treatment. Heat-treatable aluminum alloys have the best machinability in the fully aged and heat-treated tempers. Nonheat-treatable alloys machine best in the full-hard tempers.

Aluminum alloys commonly used for forging are 1100, 2014, 2017, 2218, 3003, 4032, 6061, 6151, 7075, and 7079. Welding of aluminum has progressed from an art to a science so that aluminum welding is common practice. However, because of the relatively low melting point of aluminum (1220.4°F), care must be exercised to prevent melting away an aluminum part being welded. Good welding practice also indicates the use of adequate support for the weldment, since the strength of aluminum falls rapidly as the temperature rises.

Brazing of aluminum is accomplished by furnace, dip, or torch methods. However, since the brazing temperatures are above annealing temperatures, furnace and dip-brazing will return the part to its fully annealed condition. Torch-brazed parts will be locally annealed.

For a comprehensive discussion of aluminum and aluminum alloys consult Refs. 90 and 91.

2-8.1.7.3 Magnesium[86,92,93]

Magnesium has long been recognized as the world's lightest structural metal. Alloys of magnesium are easy to machine, are easily welded, and possess excellent formability characteristics at elevated temperatures. Pure magnesium must be alloyed with other elements to provide the desired structural strength characteristics. Heat treatment further improves the properties of many magnesium alloys.

The yield strength of magnesium is defined as the stress at which the stress-strain curve deviates 0.2 percent from the modulus line since magnesium exhibits no definite yield point. The compressive yield strength of most wrought magnesium alloys is generally in the order of 70 percent of the tensile yield strength. Cast magnesium alloys tend to have equal tensile and compressive yield strengths. Generally, magnesium has good fatigue characteristics but is sensitive to stress concentrations. Notches, sharp corners, and abrupt section changes are to be avoided. Like aluminum, magnesium loses strength at elevated temperatures so that normal usage limits temperatures to about 200°F. Magnesium possesses good dent resistance as a result of its ductility; however, high aluminum content alloys tend to be subjected to stress-corrosion cracking.

The material advantages of magnesium offered by its light weight and its high strength-to-weight ratio are sometimes lost by the high initial cost per pound. This cost, however, can be recovered in production applications where the number of parts per pound of metal and the tool life economies of magnesium can be brought into the total cost picture. A comprehensive treatment of this metal and its alloys is given in Refs. 92 and 93.

2-8.1.7.4 Titanium[86,92,94]

Titanium is a light metal, about 60 percent heavier than aluminum but 45 percent lighter than steel. It is the ninth most common element and the fourth most abundant metallic element.

Titanium-base alloys are stronger than aluminum alloys and, in some respects, are superior to many steels. Their strength-to-weight ratios at elevated temperatures make them useful in environments in which other metals fail. Titanium is also the only structural metal with a corrosion-fatigue behavior in salt water equal to that in air. Titanium alloys approach the hardness of highly alloyed steels.

At the present time, titanium metal is available in 20 specific compositions which are broken down into three classes. The first class is known as the alpha alloys, which are of the highest possible purity commensurate with large-scale production. The most extensively used alpha alloy is Ti—5Al—2.5Sn, containing 5

percent aluminum and 2.5 percent tin. Alpha alloys generally have the highest strength and best oxidation resistance at elevated temperatures, but have lower strength at room temperatures and do not respond to heat treatment.

The second titanium class comprises the beta alloys. The only metastable beta alloy in commercial production today is Ti—13V—11Cr—3Al. This alloy is readily formed at room temperatures, is weldable, and can be solution-heat treated. The alloy is of a somewhat higher density than other titanium alloys and is not thermally stable above 700°F.

The third titanium class comprises the alpha-beta alloys. These vary widely in composition and general characteristics. At one extreme are highly beta-stabilized and deep-hardening alloys such as Ti—7Al—4Mo and Ti—6Al—6V—2Sn while at the other end are lean alpha-beta compositions such as Ti—6Al—4V. As a class, alpha-beta alloys have higher strength, respond to heat treatment, and are more formable than alpha alloys.

Titanium may be resistance and arc welded. The resistance welding techniques for titanium are similar to those used for stainless steel. Arc welding requires inert-gas shielding techniques. Titanium welds offer the same level of corrosion resistance as the parent metal and no stabilizing heat treatments are required.

The machining of titanium requires sharp tools, rigid set-ups, heavy feeds, slow speeds, and an ample use of coolant. Titanium's work-hardening rate is less than that of austenitic steels, copper, or alpha brasses. It is about equal to that of 0.20 percent carbon steel.

Drawing of titanium is possible; however, blanks should be profiled before forming is attempted. Several drawing stages are usually required, and an effective lubricant must be used to counteract titanium's tendency to gall.

2-8.2 NONMETALS

2-8.2.1 Strength

Typical nonmetallic materials used in the design of bodies and hulls include wood, rubber, plastics, glass, and ceramics. As a class, nonmetals have lower tensile strengths than metals. Woods have tensile strengths in the order of 6,000 psi, rubber in the order of 2,000 psi, plastics about 20,000 psi, and glass and ceramics about 7,000 psi. The strengths illustrated are approximate with each class of material having relatively strong and weak types within its class. Compressive strengths are comparable except for ceramics. Ceramics have compressive strengths approximately 10 times as great as their tensile strengths.

2-8.2.2 Weight

As a class, nonmetals tend to have a lower density than metals. Typical specific gravities for nonmetals are: wood 0.5, rubber 0.9, plastics 1.3, and glass and ceramics 2.6. The specific gravities of types within a class tend to have a significant variation. When the total vehicle weight becomes a major consideration in vehicle design, the natural tendency of the designer is to look for lighter materials to substitute for the heavier materials currently in use. One factor which makes nonmetals competitive with metals is the fact that their density is lower. Thus, in certain lightly loaded components, a material substitution can be easily made. As the strength requirements of the component are increased, the designer should also consider the strength to weight ratio of the material, the volume of the lighter part, and the cost of component material and fabrication. Many of the nonmetals compete favorably with metals in many areas.

2-8.2.3 Chemical Compatibility

Very few man-made materials are fully inert. When two dissimilar materials are placed in close proximity to each other, some interaction takes place which affects both materials. If the process is slow, taking several years to show a noticeable effect, the two materials are said to be compatible; if the process is rapid, the materials are classed as incompatible. Incompatible combinations include natural rubber and oil, natural rubber and ozone, plastics and sunlight, plastics and double-based gun propellants, and two dissimilar plastics in which the evaporating solvent in one affects the other. The determination of chemical compatibility can be a very complex process because incompatibility cannot always be predicted and usually has to be determined by empirical methods. The problem is further complicated by changes in the environment such as changing the type of fuel

used in a plastic fuel tank or by going into a hot, moist climate which accelerates the deterioration process. The chemical compatibility of nonmetals is usually more of an uncertain problem than that of metals because nonmetals encompass a larger variety of organic materials, thus significantly increasing the modes and magnitudes of chemical interaction possibilities.

2-8.2.4 Typical Nonmetals

2-8.2.4.1 Wood[95]

Although the use of wood in military vehicles has diminished, some of the properties of wood are presented here on the premise that future improvements in wood processing may again bring wood into popularity for use in military vehicles.

Woods may be classified in two groups: hardwoods and softwoods. Typical hardwoods are oak, maple, walnut, hickory, ash, and poplar; and typical softwoods are pine, cedar, fir, spruce, hemlock, and cypress. Trees are usually cut in the fall or winter when there is a minimum amount of sap in their trunks, then processed, and dried. The finished lumber retains a moisture content of about 35 percent. Although all species of wood have about the same percentage of cellulose and lignin, their mechanical properties differ considerably because of differences in their structure. In general, the mechanical properties of woods are related to their specific gravities. The higher the density of woods the higher are the values of their mechanical properties. This is shown in Table 2-3.

The moisture content, also, has a marked effect on mechanical properties; increasing moisture content up to the saturation point decreases the strength of the wood. This is illustrated in Fig. 2-4. Typical stress-strain diagrams for some of the common woods when subjected to compressive loads are shown in Fig. 2-5. Average physical properites of the most common woods are shown in Table 2-4.

Wood has an exceptionally high strength-to-weight ratio as witnessed by the use of wood in aircraft structures from early aviation days through World War II. The main drawback in those structures was the tendency of wood to absorb moisture and warp. Otherwise, wood would rival aluminum as an aircraft structural material.

Wood deterioration is usually caused by decay, insects, marine borers, or fire. Decay is caused by fungi and various bacteria. The use of preservatives such as zinc chloride or coal tar creosote decreases the rate of decay and tends to discourage borers and insects. The fire resistance of wood can be increased by surface treatment or impregnation with various fire retardant chemicals.

2-8.2.4.2 Plastics[96-100]

The term *plastics* usually refers to a class of synthetic organic material (resins) which, though solid in finished form, at some stage in their processing are fluid enough to be shaped by applications of heat and pressure. There are two basic types of plastics: (1) thermoplastics, which may be softened by applying heat and which become hard when cooled—the softening process may be repeated without the material undergoing a chemical change; and (2) thermosetting plastics, which undergo a chemical change with the application of heat and pressure— thermosetting plastics cannot be resoftened. Typical thermoplastics are ABS plastics, acetals, acrylics, cellulosics, chlorinated polyethers, fluorocarbons, polyamides, polyethylenes, polycarbonates, polypropylenes, styrenes, and vinyls. Typical thermosetting plastics include alkyds, allylics, aminos, epoxies, furanes, phenolics, polyesters, proteins, silicones, and urethanes. Typical properties of plastics are shown in Table 2-5.

As engineering materials, plastics are unique in two important respects: (1) they have good combinations of properties rather than extremes of any single property, and (2) they often permit bypassing of steps in fabrication in design. When compared with metals, properties that make plastics more favorable are:

a. Better chemical and moisture resistance
b. Better resistance to shock and vibrations
c. Transparent or translucent
d. Quieter—tend to absorb sound vibration
e. High abrasion and wear resistance
f. Self-lubricating
g. Often easier and less costly to fabricate

Properties of plastics that may be unfavorable in particular applications are:

TABLE 2-3 RELATION OF STRENGTH OF WOOD TO SPECIFIC GRAVITY

Property	Moisture Content	
	Green	Air-dry (12% moisture)
Static bending:		
Fiber stress at prop. limit, psi	$10{,}200G^{1.25}$	$16{,}700G^{1.25}$
Modulus of rupture, psi	$17{,}600G^{1.25}$	$26{,}200G^{1.25}$
Modulus of elasticity, psi	$2.36 \times 10^6 G$	$2.8 \times 10^6 G$
Compression parallel to grain:		
Fiber stress at prop. limit, psi	$5{,}250G$	$8{,}750G$
Stress at ultimate, psi	$6{,}730G$	$12{,}200G$
Modulus of elasticity, psi	$2.91 \times 10^6 G$	$3.38 \times 10^6 G$
Compression perpendicular to grain:		
Fiber stress at prop. limit, psi	$3{,}000G^{2.25}$	$4{,}630G^{2.25}$

Note: G = the specific gravity of the wood, lb/in.^{3}

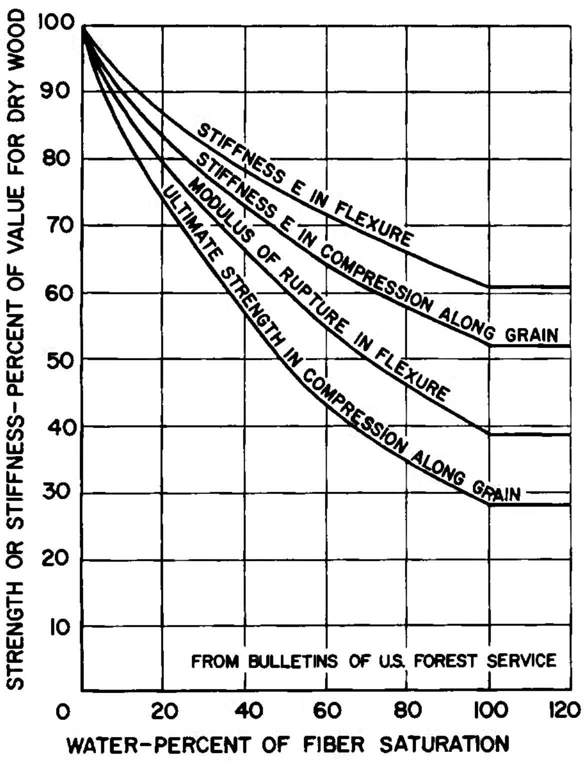

Figure 2-4. Effect of Moisture on Strength and Stiffness of Softwoods

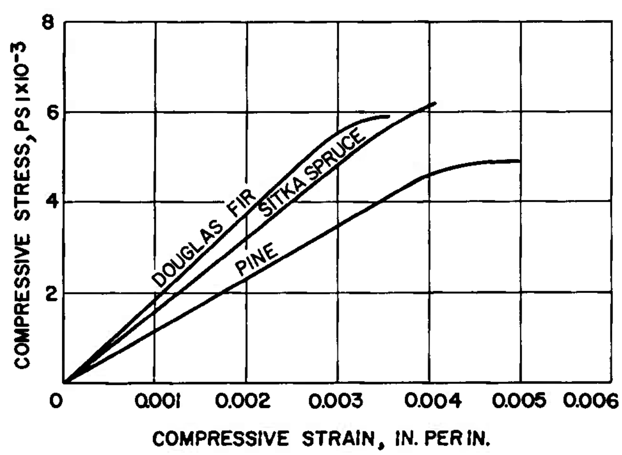

Figure 2-5. Typical Stress-strain Curves for Wood in Compression Parallel to to Grain

a. Lower strength

b. Much higher thermal expansion

c. More susceptible to creep, cold flow, and deformation under load

d. Lower heat resistance

e. More subject to embrittlement at low temperatures

f. Softer

g. Less ductile

h. Change dimensions through absorption of moisture or solvents

TABLE 2-4 AVERAGE PHYSICAL PROPERTIES OF WOOD*

		Bending			*Compression Parallel to Grain*			
Commercial Name	*Specific Gravity*[a]	*Prop. Limit,* psi $\times 10^{-3}$	*Modulus of Rupture,* psi $\times 10^{-3}$	*Modulus of Elasticity,* psi $\times 10^{-6}$	*Prop. Limit,* psi $\times 10^{-3}$	*Ult. Str.,* psi $\times 10^{-3}$	*Prop. limit in compression perp. to grain,* psi $\times 10^{-3}$	*Ult. shear strength parallel to grain,* psi $\times 10^{-3}$
Ash, commercial white	0.58	8.9	14.6	1.68	5.58	7.28	1.51	1.92
Cedar, eastern red	0.47	3.8	8.8	0.88	—	6.02	1.14	—
Cypress, southern	0.46	7.2	10.6	1.44	4.74	6.36	0.90	1.00
Douglas fir, Rocky Mt.	0.43	6.3	9.6	1.40	4.66	6.06	0.82	1.07
Elm, American	0.50	7.6	11.8	1.34	4.03	5.52	0.85	1.51
Hemlock, eastern	0.40	6.1	8.9	1.20	4.02	5.41	0.80	1.06
Hickory, true	0.73	10.9	19.7	2.18	—	8.97	2.31	2.14
Maple, red	0.54	8.7	13.4	1.64	4.65	6.54	1.24	1.85
Oak, red	0.63	8.4	14.4	1.81	4.61	6.92	1.26	1.83
Oak, white	0.67	7.9	13.9	1.62	4.35	7.04	1.41	1.89
Pine, northern white	0.36	6.0	8.8	1.28	3.68	4.84	0.55	0.86
Pine, southern yellow:								
Longleaf	0.58	9.3	14.7	1.99	6.15	8.44	1.19	1.50
Shortleaf	0.51	7.7	12.8	1.76	5.09	7.07	1.00	1.31
Redwood (virgin)	0.40	6.9	10.0	1.34	4.56	6.15	0.86	0.94
Spruce, Sitka	0.40	6.7	10.2	1.57	4.78	5.61	0.71	1.15
Tamarack	0.53	8.0	11.6	1.64	4.78	7.16	0.99	1.28

*Results of tests on small[b], clear specimens under air-dry conditions with 12 percent moisture content as reported in Ref. 95.

a Based on weight when overdry, and volume at 12 percent moisture content.

b Test specimens 2 X 2 in. in cross section. Bending specimens 30 in. long; other specimens shorter depending upon kind of test.

TABLE 2-5 PROPERTIES OF PLASTICS[98]

Classification	*Useful Temperature Range, °F*	*Specific Gravity*	*Thermal Expansion, 10^{-6}in./in./°F*	*Thermal Conductivity, Btu/hr/ft²/°F/in.*	*Resistivity, ohm-cm*	*Dielectric Strength (short time), v/mil*	*Ultimate Tensile Strength 1,000 psi*	*Modulus of Elasticity, 10^5 psi*
Acetals	−50 to 150	1.4	45	0.2	4^{14}	500	10	4
Acrylics	−400 to 175	1.1 to 1.2	30 to 60	0.2	10^{14} to 10^{16}	400 to 530	6 to 11	2.5 to 5
Cellulosics	−10 to 110	1.1 to 1.3	45 to 110	0.1 to 0.2	10^{6} to 10^{15}	250 to 600	1.5 to 8.5	0.5 to 4
Chlorinated Polyethers	−10 to 225	1.4	45	0.8	10^{16}	400	6	2
Fluorocarbons	−400 to 400	2.1 to 2.3	40 to 105	0.1 to 0.4	10^{18} to 10^{19}	400 to 600	2.5 to 5.5	0.5 to 3
Nylons	0 to 200	1.1	45 to 70	0.1	10^{13} to 10^{15}	390 to 500	7 to 12	2 to 4
Polycarbonates	−150 to 250	1.2	35	0.2	3^{17}	400	9 to 11	—
Polyethylenes	−75 to 150	0.9	85 to 165	0.2	10^{15} to 10^{19}	500	1 to 4.5	0.5 to 2
Polypropylenes	0 to 210	0.8	65	0.1	5^{17}	750 to 850	5	1 to 2.5
Styrenes	−25 to 175	1.1	25 to 55	0.1	10^{13} to 10^{19}	300 to 650	3 to 8	1 to 5
Vinyls	0 to 120	1.1 to 1.7	25 to 130	0.2	10^{11} to 10^{16}	15 to 1400	1 to 9	0.5 to 4
Alkyds	300 to 350	2.0 to 2.4	10 to 30	0.2 to 0.6	10^{14}	300 to 350	3 to 10	5 to 25
Allytics	300 to 450	1.3 to 1.9	10 to 60	1.2 to 2.2	10^{5} to 10^{16}	350 to 450	4 to 7	4 to 20
Aminos	200 to 400	1.4 to 2.0	10 to 30	0.2 to 0.4	10^{9} to 10^{14}	125 to 400	5 to 10	10 to 19
Epoxies	100 to 400	1.0 to 2.1	15 to 50	0.1 to 0.7	10^{9} to 10^{16}	350 to 550	0 to 30	1 to 3.5
Phenolics	200 to 450	1.2 to 3.0	10 to 65	0.1 to 0.4	10^{6} to 10^{13}	75 to 425	2.5 to 10	4 to 50
Polyesters	200 to 300	1.1 to 1.4	25 to 55	0.1	10^{12} to 10^{13}	250 to 570	1 to 10	2 to 7
Silicons	600 to 700	1.7 to 2.0	5 to 30	0.1	3^{14}	250 to 400	4 to 5	25 to 30
Urethanes	—	1.3	50 to 100	1.4	2^{12}	450 to 700	5 to 8	1

i. Flammable

j. Some varieties may be degraded by sun or other ultraviolet radiation

k. Most plastics cost more than metals per cubic inch; nearly all cost more per pound

l. Some types support certain fungus growth

As in any other generality there are exceptions to the rule, so that the above mentioned attributes and disadvantages should be applied with care to any design selection of plastics or metals.

Typical uses of plastics on bodies and hulls of military vehicles include armor, ducts, guides and wear surfaces, housings, boxes and containers, electrical equipment, insulation, weather closures, and liners. As the plastic technology progresses, it is highly probable that entire bodies and hulls may be made from plastic. As a forerunner, the XM150, 1000-gallon, 6 X 6, Water Tank Truck[101] utilized a plastic tank with some degree of success. The test of the vehicle indicated that the plastic tank could maintain water at a nearly constant temperature at both high and low ambient temperatures without external means.

2-8.2.4.3 Rubber[102,103,117,118]

Natural rubber is obtained from the *hevea brasiliensis* tree of the Far East in the form of latex. The latex is coagulated, washed, rolled, dried, and smoked to form crude rubber. Crude rubber has a number of undesirable qualities, the principal of which is lack of strength. This is improved through vulcanization by the addition of sulfur at about 160°F. The degree of vulcanization affects the properties of the rubber. Rubber containing a high percentage of sulfur is hard, while a low sulfur content results in a soft rubber. Typical stress-strain curves for natural rubber are illustrated in Fig. 2-6.

Natural rubber has good energy-absorbing characteristics since large deformations can be produced before fracture. In addition, it has good tear and wear resistance properties and low electrical conductivity. Natural rubber does not perform well, however, when exposed to chemicals and petroleum derivatives, nor is it recommended for applications where resistance to sunlight, ozone, oxygen, or heat is a major factor.

Various synthetic rubbers have been developed from substances such as coal, butadiene, styrene, and polysulfides. Some of the synthetics, such as polyisoprene, closely resemble natural rubber in chemical composition and physical properties.

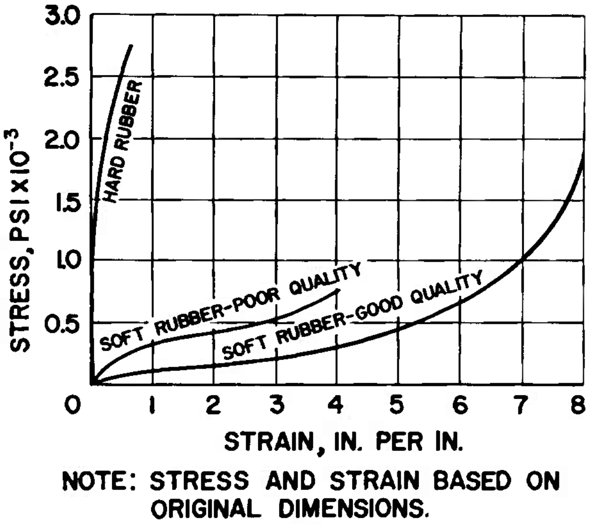

Figure 2-6. Typical Stress-strain Curves for Natural Rubber

Ref. 103 gives a comparative tabulation of 41 properites of 14 different elastomers including natural rubber. A more comprehensive discussion of rubber and rubberlike materials is given in Ref. 102.

2-9 CORROSION CONTROL[104-107]

2-9.1 DEFINITION

Corrosion may be defined as the gradual deterioration or disintergration of materials by direct chemical or electrochemical reaction with its environment. It is found in both metals and nonmetals; although corrosion in metals is the most common and is, therefore, the most understood. The corrosion effects may be localized in the form of pitting of the surface, or they may be a uniform deterioration of the surface. Many of our most useful metals undergo a slow chemical change when exposed to air, water, gases, acids, or salt solutions. Such action results first in the formation of a surface coating which causes the metal to lose its characteristic

metallic luster. As the action continues, the coating often breaks away, permitting the corrosive action to penetrate deeper into the metal until the corroded part becomes unfit for further service. The corrosion may penetrate to a sufficient depth to reduce the effective cross section of a stressed component, thus causing it to fail at a load far below that which it is otherwise capable of carrying. Furthermore, the corroded surface is roughly pitted, and the bottom of each pit is an effective stress concentration point which serves to reduce appreciably the fatigue life of the material.

Fungus decomposition is another form of material deterioration that primarily affects nonmetals. Fungi are a lower form of plant life that gain sustenance either by decomposing nonliving organic substances or by attacking living plants or animals. Like all other living organisms, fungi require moisture for their growth; and thus, the relative humidity of the atmosphere exerts an extremely important influence over them. Most forms require a relative humidity of 70 to 100 percent and temperatures above 50°F. They are capable of bringing about the oxidation, reduction, or hydrolysis of many kinds of organic compounds. Most fungi require oxygen but do not require light for growth. They can create serious problems, especially in tropical areas of the world where the most favorable climatic conditions for their growth are found. Fungi attack untreated materials such as fabric covers, electrical insulations, glued joints, wood, paper, plastics, and leather. They accelerate the corrosion of metals by absorbing water and retaining it on the surface of the metal. Many fungi are electrical conductors and cause functional failures in electronic equipment by creating short circuits. In designing equipment using fungus susceptible materials, protection should be provided by incorporating suitable fungicides into the materials by using volatile fungicides, using non-nutrient materials, by applying moistureproof and fungusproof coatings, or by reducing the humidity in the environment of the equipment to a zone where fungus growth will not take place.

2-9.2 FACTORS STIMULATING CORROSION

Most metals are not inherently stable, so that environmental factors tend to cause them to revert to their natural metallic ore states. Iron, the most useful of metals, is seldom found in the pure state in nature because its chemical activity causes it to corrode rapidly into a hydrated oxide. Every metal has a tendency to dissove in water. When this occurs, an equivalent amount of hydrogen is liberated. Any condition that promotes the formation of hydrogen favors corrosion of the metal. Thus, the solution of oxygen or of an acid in water promotes corrosion, which in turn yields corrosion products which consist of oxides, carbonates, and sulfides.

While corrosion of metals is caused chiefly by the action of oxygen, water, and carbon dioxide, none of these agents alone is capable of corroding iron. Dry air and pure water, which contains no dissolved gases, have no effect upon the metal; but when exposed to both air and moisture, the metal corrodes rapidly. Water that contains dissolved carbon dioxide attacks such metals as iron and lead quite readily. Furthermore, the presence in the air of gases containing sulfur, such as hydrogen sulfide or sulfur dioxide, also causes some metals to corrode quite rapidly.

Atmospheric corrosion of metals is stimulated by a high concentration of moisture in the air. This serves to maintain a film of water on the surface of the metal. Acids, acid gases, and sulfur compounds that occur in the air tend to dissolve in, or react with, this moisture to form an acid film on the exposed surfaces. Thus, the stage for corrosion is set.

Other factors which influence corrosion are: the rate of flow of the corrosive fluid in contact with the metal, the inherent corrosion properties of the metal, the ambient temperature, stresses in the material due to external loads or from other sources, and contact with metals that are dissimilar as referenced to the electromotive scale. This last factor is discussed in greater detail in par. 2-9.4.

2-9.3 FACTORS INHIBITING CORROSION

Practices aimed at inhibiting corrosion include the use of sacrificial metals such as zinc, aluminum, or magnesium to protect iron or steel; the use of pure aluminum to protect copper-aluminum alloys; the use of protective coatings such as paints, surface plating, metalizing, or chemical corrosion inhibitors; the

treatment of metals to render them insoluble in acids; and the neutralization of the corrosive environment where this is possible. Metals of high purity generally resist attack by corrosion better than low purity metals, because electrolytic degradation takes place around the impurities to produce cells of corrosion which then spread through the metal. Alloys are generally less resistant than are pure metals; although the addition of aluminum or nickel to iron generally improves corrosion resistance. Perhaps the most successful employment of the corrosion resistance of pure metals in improving the corrosion properties of an alloy has been the application of high purity aluminum to copper-aluminum alloys, resulting in the popular laminated material known as Alclad. This has made possible the use of high-strength aluminum alloys without danger of deterioration by corrosion.

2-9.4 ELECTROCHEMICAL (GALVANIC) CORROSION [104]

Electrochemical or galvanic corrosion occurs when two dissimilar metals are in contact with each other while submerged in, or coated by, a suitable electrolyte. Moist air, particularly if it contains the usual sulfides and carbonates normally found in air, is a satisfactory electrolyte for the purpose of galvanic corrosion. Under these conditions, a difference of electrical potential will exist between the two metals causing a minute electrical current to flow between them. This produces a plating action resulting in a loss of metal that is approximately proportional to the galvanic current—the metal higher (less noble) in the electromotive series going into solution. This type of corrosion can be minimized by employing the following principles:

a. Using materials that are as close to each other as possible on the electromotive scale

b. Insulating the materials from each other

c. Protecting both metals from contacting each other by applying a suitable paint between them

d. Avoiding designs that result in relatively large areas of the more noble metal and small areas of the less noble metal

e. Increasing the thickness of the less noble metal

2-9.5 ELECTROLYTIC CORROSION [85]

Electrolytic corrosion is generally caused by the leakage of current from electrical circuits. Galvanic corrosion is a special case in which the electrical current is generated by the galvanic action of the structural components. Stray currents of feeble intensity may accelerate corrosion and may also cause corrosion at points far removed from the source of the electrical leakage. This type of corrosion may be minimized by careful attention to shielding and insulation of electrical lines and leads.

2-9.6 METHODS OF MINIMIZING CORROSION [119]

The corrosion behavior of specific metals and alloy groups is given in Ref. 107. While corrosion control is a particular function of the material and its environment, some general methods of corrosion control are given in the paragraphs that follow.

2-9.6.1 Protective Metal Coating [104]

Metals can be given a coating of corrosion resistant metal to protect the basic material. Typical coating materials are zinc, cadmium, tin, lead, nickel, copper, and chromium. The coating used depends upon the metal to be coated, the corrosive environment, and the cost of coating. The desirable properties of metallic protective coatings include: durability, ease of application and low cost, good adhesion to the base metal, uniform density and freedom from pinholes, wearability, and, at times, appearance. The coating should be neutral to the base metal or somewhat anodic. Since no one coating offers all of the desirable properties, an analysis is required to obtain the best corrosive coating for each particular application. (See MIL-A-8625 [anodizing], MIL-E-16400 and QQ-C-320 [chromium], QQP-416 [cadmium], QQ-E-325 [zinc], QQ-S-365 [silver], MIL-L-13762 and MIL-L-13808 [lead], MIL-T-10727 [tin], MIL-M-3171 [magnesium], QQ-C-525 and QQ-C-576 [copper], and QQ-N-290 and MIL-P-18317 [nickel].)

2-9.6.2 Oxide or Phosphate Coating [104]

Oxide coatings (see MIL-F-13924 and MIL-F-495) are produced by heating the metal

in various media, depending upon the color or character of coating desired. These coatings are usually cathodic to the metal and offer only a limited protection against corrosion.

Phosphate coatings (see MIL-P-16232) are applied by immersing the metal in a dilute solution of warm phosphoric acid which forms a crystalline phosphate coating on the surface of the metal. In general, these coatings are slightly less cathodic than oxide coatings, and, therefore, their protection capabilities are limited to the extent that further protection is required if the metal is to be exposed to a corrosive environment.

2-9.6.3 Protective Paints [104]

The application and maintenance of painting coatings (see MIL-STD-193) is by far the most generally used method of preventing corrosion due to the atmosphere. Included as "paints" are varnishes, lacquers, and enamels. In general, the paints form a barrier of film-forming materials which, together with the pigment in the paint, help to exclude moisture from the surface of the metal. Plasticizers are sometimes added to the paints to give flexibility to certain film-forming materials that tend to become rigid when dry.

The selection of the best protective paint depends upon the material to be painted, the environment to which it is to be exposed, and the current paint technology. In general, the material to be painted should be thoroughly cleaned and dried. Some metallic surfaces must also be treated to allow the paint to adhere to the surface. A priming coat should be applied, in a dry atmosphere, at temperatures above 50°F (10°C). The priming coat should have sufficient wetability to fill and wet completely any depressions in the metal surface. Too thick a prime coat will bridge the small depressions and leave part of the surface underprotected. Too thin a prime coat will leave thin spots at the surface ridges, thus also leaving part of the surface underprotected. The finish coat can be added by spraying or brushing; the former is preferred for large surfaces since it has the advantage of speed of application. If properly done, both methods offer the same degree of protection.

Some basic considerations that may influence the selection of paint pigments are the following:

a. The basic substances that inhibit corrosion are litharge, red lead, blue lead, white lead, zinc oxide, and zinc dust.

b. Chrome compounds—such as basic lead chromate, normal lead chromate, and zinc chromate—are useful in corrosion prevention.

c. Neutral substances that do not ionize to form acid reactions are considered inert. Such substances include black, brown, and red oxides of iron, china clay, silica, talc, and barium sulfate.

d. Substances that form a galvanic couple with the coated material cause rapid corrosion in the presence of moisture.

2-9.6.4 Surface Passivation [104,107]

Passivity is the property by which metals become inactive in a specific environment. A metal may be made passive by a surface treatment of the metal or by the addition of inhibitors. For instance, stainless (18-8) steel is not significantly less active than plain carbon steel in its untreated state; however, treatment by immersing the polished steel in nitric acid forms an oxide film which renders the steel passive. In the passive state, steel is one of the most noble metals. This surface film, however, can be destroyed by exposing the steel to a reducing atmosphere.

Aluminum is made passive by immersion in a chromic acid solution under specific voltage and temperature conditions (anodized). Anodizing of aluminum can also be accomplished by using sulfuric acid to form a protective coat.

Passivation of other metals meets with various degrees of success, depending upon the metal. Magnesium, for instance, can be anodized[107]; however, the degree of protection afforded by this process is less than when used with aluminum.

2-10 SUPPLEMENTARY DOCUMENTS THAT INFLUENCE DESIGN

A number of documents exist that have a regulatory influence on vehicle design and should be familiar to the designer. Many of these in the form of Army Regulations, Military Specifications, Military Standards, and other

official directives are referenced in the applicable portions of this handbook. This paragraph presents some supplemental documents that contain useful background information and design guidance on recommended practices in specific areas of design.

2-10.1 MILITARY CHARACTERISTICS (MC's)

The document known as the Military Characteristics (MC's) is prepared for each new vehicle to be developed, and is the basic document to the designer. It contains five sections. Section I contains a general statement of the requirement or materiel required; operational and organizational concepts for the materiel to be developed; a statement of Navy, Air Force, Marine Corps, British, and Canadian interests; feasibility of development which delineates the action to be taken if problem areas or a need for the use of critical materials arise; and background information such as data based on past experience, related equipment, and vehicle usage.

Section II covers the operational characteristics of the equipment to be developed. It defines and describes the configuration of the equipment, its performance characteristics, its durability and reliability capabilities, and its transportability requirements.

Section III deals with special characteristics such as requirements for environment and terrain capabilities, CBR resistance kits, maintenance and reliability, and human factors engineering.

Section IV deals with the establishment of an order of priority for the major characteristics.

Section V lists the items which are intended to be superseded by the equipment which is to be developed.

2-10.2 CODE OF FEDERAL REGULATIONS (CFR) TITLE 49—TRANSPORTATION

The *Code of Federation Regulations (CFR) Title 49—Transportation* contains Federal rules and regulations on transportation. Compliance with the CFR is mandatory for civilian vehicles operating on Federal highways or crossing state lines, and is optional for military vehicles. Although compliance with the CFR is not mandatory for military vehicles, it does provide useful guidance in the practical design of safe vehicles. Of particular interest to the vehicle designer are Parts 71 to 90 and Parts 190 to 197 of the CFR. Parts 71 to 90 give general information and design requirements pertaining to shipping containers and to vehicles used in the transportation of explosives and other dangerous cargo. These parts are particularly applicable to the design of tank-type bodies for the transportation of liquid fuels, liquefied gases, and corrosive liquids. Parts 190-197 are familiar to many under the title, "Motor Carrier Safety Regulations of the Interstate Commerce Commission". Part 193 is particularly applicable to vehicle design; it lists and defines the various vehicle components and accessories necessary for safe vehicle operation on roads.

The *CFR Title 49—Transportation* Parts 71 to 90 and Parts 190-197 can be obtained from the Superintendent of Documents, U. S. Government Printing Office, Washington, D. C. 20402. Parts 190-197 can also be obtained from The American Trucking Associations, Inc., 1616 P Street, Northwest, Washington, D. C. 20036.

2-10.3 STATE HIGHWAY REGULATIONS

Various statutes have been established by the separate state governments regulating vehicle sizes, weights, axle and wheel loadings, tire requirements, vehicle lighting, inspection requirements, and safety provisions. These regulations vary with each state and are subject to frequent revision which makes their compilation difficult and requires frequent updating. However, as a service to the trucking industry, these data are compiled annually and published in April of each year in the *Fleet Reference Annual* edition of the *Commercial Car Journal.* Copies can be obtained from the Chilton Company, Chestnut and 56th Streets, Philadelphia, Pa. 19139.

2-10.4 SAE HANDBOOK

The *SAE Handbook* contains civilian (and some military) standards, information reports, recommended practices, test procedures, and design data on metals, nonmetallic materials, threads, fasteners, common parts, electrical

equipment, lighting, powerplant components and accessories, passenger cars, trucks, busses, tractors, earth-moving equipment, and on marine equipment. While all parts of the Handbook may not be directly applicable to the design of military vehicles, the Handbook serves as a useful source for design data and background information and is often listed as a reference document in Military Standards for vehicles.

The *SAE Handbook* can be obtained from The Society of Automotive Engineers, Inc., Department S6, 485 Lexington Avenue, New York, N. Y. 10017.

2-10.5 ASME BOILER AND PRESSURE VESSEL CODE

The *ASME Boiler and Pressure Vessel Code* is a reference document used in the design of some military vehicles, mainly vehicles with tank bodies. The code consists of nine sections of which Section II, "Material Specifications"; Section VIII, "Unfired Pressure Vessels"; and Section IX, "Welding Qualifications" are of interest to the vehicle designer. Section II covers materials which are qualified to meet Code requirements and lists the chemical compositions and allowable working stress levels for ferrous and nonferrous materials. The working stresses allowed by the Code are conservative, and judgment should be used in their application in order to avoid undue weight penalties. Section VIII covers minimum construction requirements for the design, fabrication, inspection, and certification of unfired pressure vessels. The section covers welded, riveted, forged, and brazed fabrication methods using carbon, low- and high-alloy steels, cast iron, nonferrous metals, and clad and lined materials. Section IX presents the rules governing the qualification of welding procedures, welding equipment, and welding operators for all types of manual and machine arc and gas welding processes permitted in other sections of the Coode.

The *ASME Boiler and Pressure Vessel Code* can be obtained from the American Society of Mechanical Engineers, United Engineering Center, 345 E. 47th St., New York, N. Y. 10017.

2-10.6 MIL-HDBK-5A

MIL-HDBK-5A, *Metallic Materials and Elements for Flight Vehicle Structures*, is widely used in the aerospace and military automotive industries as a reference document. The handbook contains a comprehensive compilation of material properties, structural design information, and miscellaneous design data not readily available elsewhere. Its purpose is to provide a single source of uniform, accurate data, and eliminate the need for consulting many references. Many of the data in the handbook are empirically derived; but are, in general, more realistic than similar theoretical data presented in many textbooks.

MIL-HDBK-5A can be obtained from the Superintendent of Documents, U. S. Government Printing Office, Washington, D. C. 20402.

2-11 STANDARDIZATION

The Department of Defense Standardization Program was authorized by Public Law 436 of the 82nd Congress and by Public Law 1028 of the 84th Congress and was implemented by *DOD Standardization Manual M201* and AR 1-70, *Standardization Among Armies of United States, United Kingdom, and Canada*, and (CM) AR 1-71, *Standardization Among the Countries of the North Atlantic Treaty Organization (U)*. The purpose of the program is to improve the efficiency and effectiveness of logistical support to, and the operational readiness of, the Armed Forces, and to conserve money, manpower, time, production facilities, and natural resources of the nation. The means by which these goals can be attained are:

a. By adopting the minimum number of sizes, kinds, and types of items and services essential to military operations

b. By providing the greatest practical degree of component interchangeability

c. By developing standard terminology, codes, and practices

d. By preparing engineering and purchasing documents that provide for the design, purchase, and delivery of items that are consistent with the objectives of the Department of Defense Standardization Program

e. By supplying the military departments with the most reliable equipment possible through the use of materiel that has been evaluated in accordance with established Government specifications and standards.

In order for this program to be successful, standardization must be applied at all stages of the design process. Wherever it is practical to do so without jeopardizing or degrading the functional objectives of the design, maximum employment should be made of standard parts, components, and subassemblies. A key factor in reducing the overall and long-range costs of logistical support is to design so as to standardize for both physical and functional interchangeability. All parts that will be subject to removal and replacement should be standard and uniform. Designs must not be based upon the use of components that require special manufacture, or that are produced by a single manufacturer when equivalent parts are available from several other manufacturers. Complete mechanical and electrical interchangeability should be provided for among like assemblies, subassemblies, and replacement parts. Maximum use should be made of parts already listed in the *Army Repair Parts List* or in the *Federal Supply Catalog* and of commercial, nonproprietary parts. Designs should also be controlled to permit free use of standard common tools and general test equipment and tools and test equipment already listed in the *Army Stock List*.

Standard assemblies, subassemblies, components, parts, and miscellaneous standard hardware applicable to tank-automotive materiel are catloged in the *Standard Military Component Directory*, Vols. I, II, III, and IV, compiled and issued by the Standards Section, AMSTA–RSS.2, Standardization Branch, Procurement Engineering Division, U. S. Army Tank-Automotive Command, Warren, Michigan 48090.. Items listed in this Directory are to be utilized to the maximum practical extent commensurate with the objectives of the new design. The full use of this publication is a prime factor in the DOD Standardization Program.

In addition to the catalog of standard components, the publication contains an index of those International Standard Agreements that affect tank-automotive design. Where these impose limitations on the design, it is imperative that early consideration be given to these agreements. Where noncompliance is indicated, prescribed steps must be taken by the U. S. to assure the cognizance of others that are members of the agreements.

SECTION III—CONSIDERATIONS FOR HULL-TYPE VEHICLES

2-12 RELATIONSHIP OF HULL TO OTHER ELEMENTS

2-12.1 GENERAL

Normally, tracked vehicles have hulls rather than bodies, as discussed in par. 1-1.3. In brief, hulls serve as housings for the power plant, crew, cargo, and equipment; as watertight compartments for amphibious, fording, and floating operations; and as protective shells for the crew, cargo, and equipment against the weather, projectiles, missiles, and CBR agents. The hull must be strong enough to withstand the applied loads, stiff enough to resist deformations, flexible enough to yield to shocks and impact loads, large enough to house the crew, weapons, cargo and equipment, and light enough and small enough to maximize maneuverability and permit use in airborne operations.

The hull designer is usually in the position of having to compromise several design areas in order to achieve a reasonable balance in the overall vehicle design. The degree of compromise depends, to a large extent, on the importance he places on each area; so that the end product is strongly dependent upon his capabilities and experiences. As background information to assist the hull designer, this section points out some of the relationships the hull bears to other elements of the vehicle and which the designer must consider when developing the hull design.

2-12.2 HULL-SUSPENSION RELATIONSHIPS

The suspension system of a vehicle is that complex of mechanical, structural, pneumatic,

hydraulic, and electrical components whose functions are directly or indirectly associated with the provision of an elastic support for the hull or body. Since it is the only connection between the hull and the ground, it must also transmit the driving and braking forces, steering forces, lateral and longitudinal stabilizing forces, weapon recoil and counterrecoil forces, and forces resulting from the operation of vehicle-mounted equipment such as cranes, shovels, draglines, and dozer blades. A comprehensive discussion of vehicle suspension systems is given in Ref. 3.

The primary relationship of the hull to the suspension system is to provide suitable anchor points, or mountings, for the various suspension components. These are in the form of pads, bosses, brackets, ears, studs, and openings—integral with the hull structure—that are designed to safely resist and transmit the forces previously mentioned (see Fig. 2-7). Some of the major suspension components for which mountings must be provided on the hull are the torsion bar assemblies, road wheel arm support housings, shock absorber brackets, road wheel arm bumper stops, track support roller brackets, idler wheel supports, and the track tensioning idler supports.

Figs. 2-8 and 2-9 show two methods of attaching road wheel arm support housings to the hull—the former using bolts and the latter using weld. The hull in Fig. 2-8 is shown upside down for clarity. Note, also, the mounting of the road wheel arm bumper stops. Welded construction has the advantage of permanence and eliminates the need for providing seals against dirt and water, but its permanence makes the replacement of a damaged housing extremely difficult.

The torsion bars generally extend completely across the width of the hull, crossing just above the belly plates. Being thus placed inside the hull, they are protected from damage by road hazards, rocks, logs, small arms projectiles, and—to some extent—from land mines. They must be protected from damage on the inside of the hull by being boxed in with selectively placed structured reinforcing members or by suitable covers. Furthermore, since the torsion bars extend completely across the width of the hull, corresponding road wheel support housings on opposite sides of the vehicle must be displaced longitudinally with respect to each other to permit a side-by-side placement of the torsion bars. This is shown in Fig. 2-10.

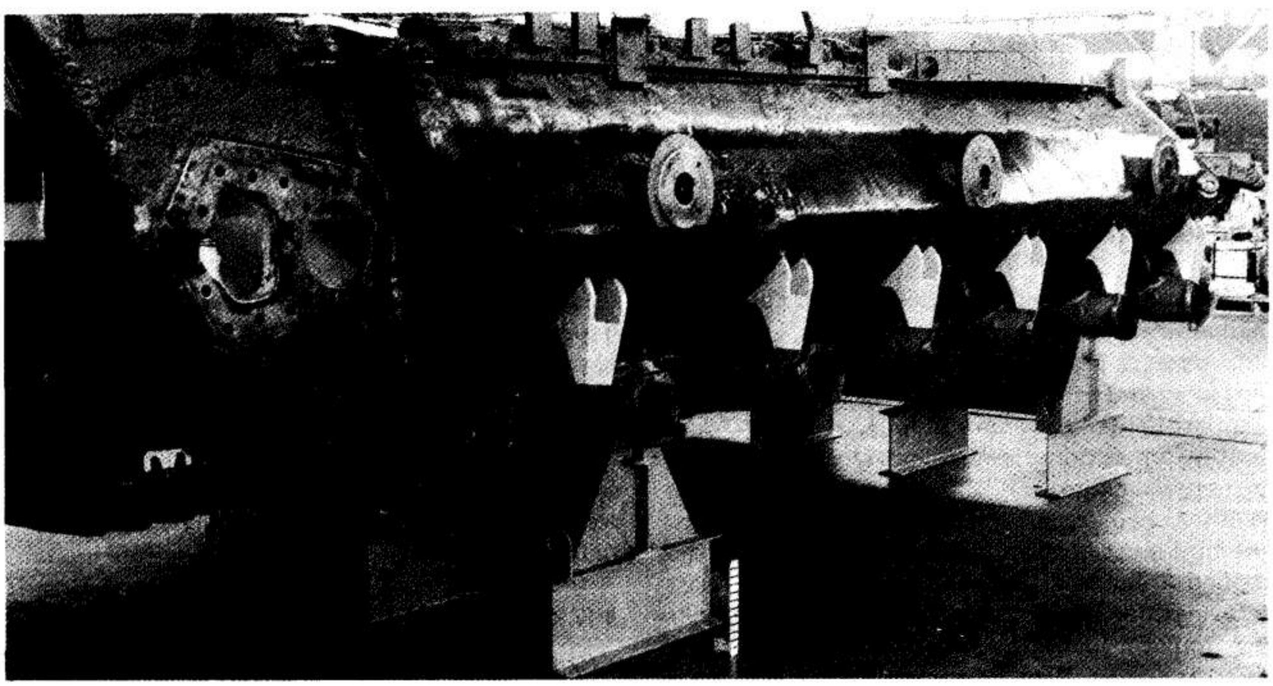

Figure 2-7. M48 Tank Hull Showing Suspension Housings and Brackets

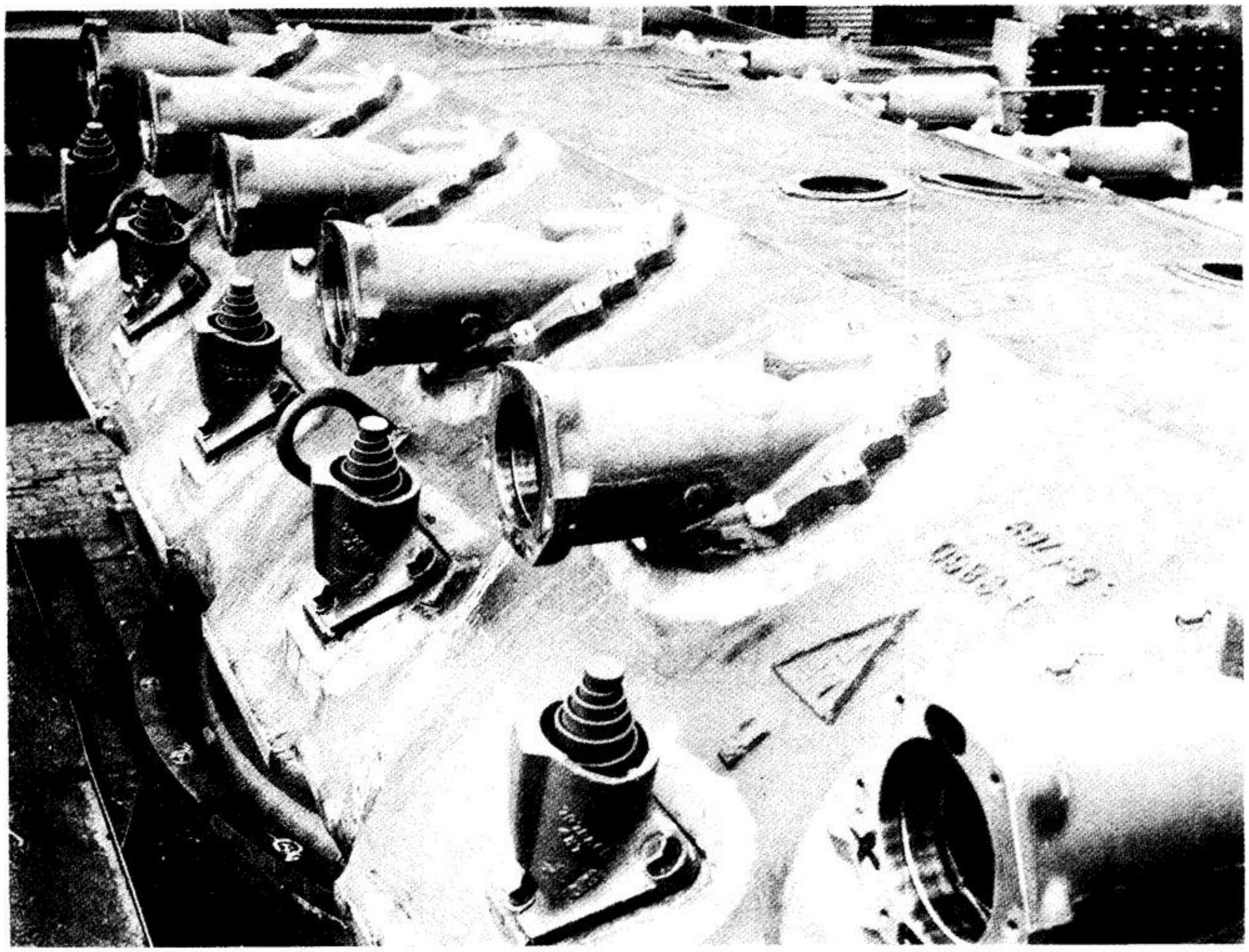

Figure 2-8. Medium Tank Hull With Road Wheel Arm Support Housings Bolted to Hull

2-12.3 HULL-TURRET RELATIONSHIPS

The turret of a combat vehicle is a rotatable structure, located on top of the hull, which contains the vehicle's main armament and often its secondary armament as well. It is usually a dome-shaped fully enclosed armored structure, particularly as found on tanks and tank-type vehicles. Being fully rotatable, it provides the tank with the capability of rapidly traversing its weapons through a full 360° arc without the need to pivot the vehicle.

A cylindrical basket structure (turret basket), which is a part of the turret assembly, extends into the hull to a point just above the hull floor and rotates with the turret as a single unit. This basket provides physical support for the turret crew and storage space for various items such as racks for ready rounds for the main weapon, boxes of ammunition for the secondary weapons, spare periscopes, a tray of oddments, and miscellaneous necessary equipment. The principal reason for having the basket rotate with the turret is to facilitate the movements of the crew and give ready access to controls and ammunition. Although the loader rides with the turret basket, his position need not be indexed to the ammunition storage compartment in the hull until the rounds carried in the turret basket ready racks are expended.

At the present time, the tank armament normally requires the services of three men for rapid, effective operation. These are the tank commander, who locates targets, directs the fire, and otherwise takes charge of the tank's operations; the gunner, who aims and fires the armament; and the loader, who loads the weapon, extracts the spent cartridge cases, and

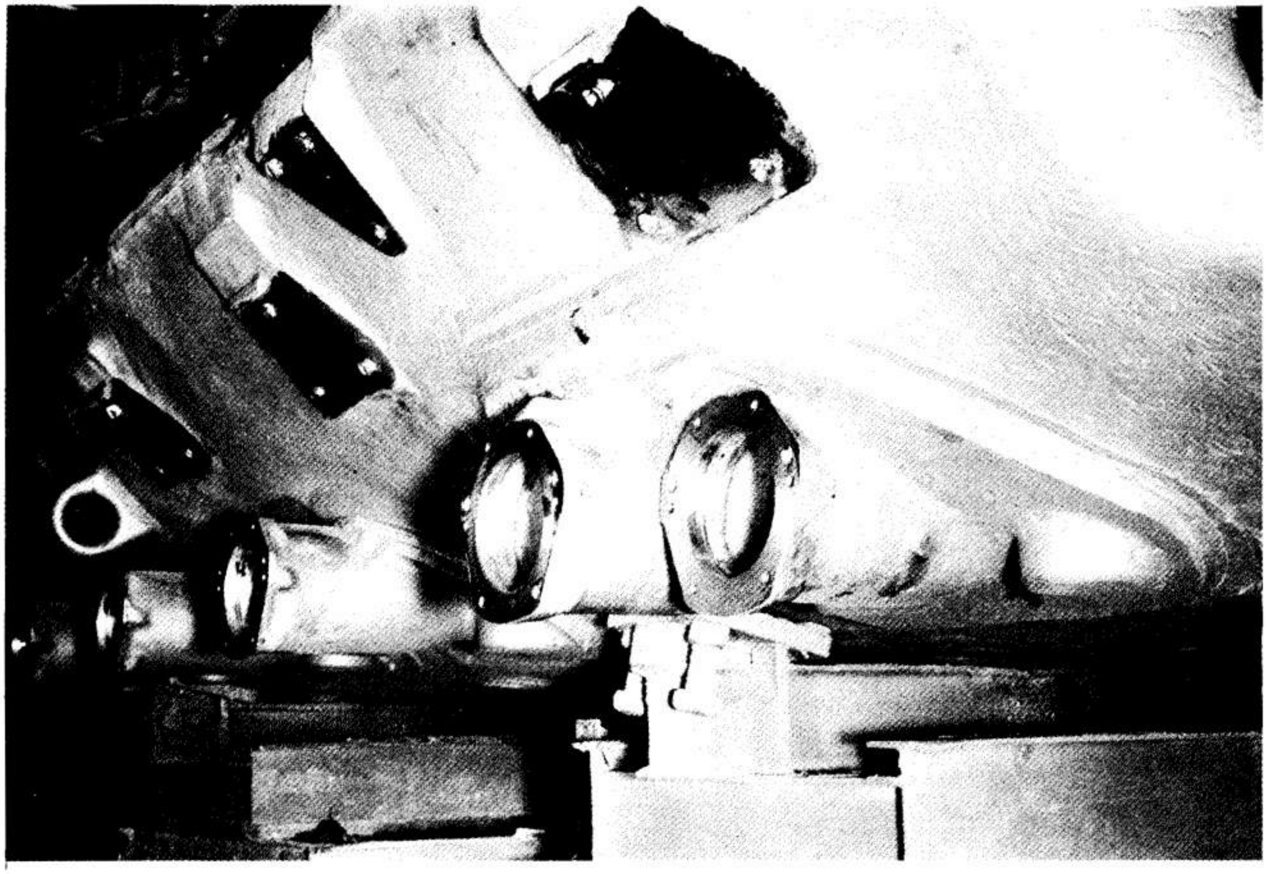

Figure 2-9. Medium Tank Hull With Road Wheel Support Housings Welded to Hull

otherwise assists in the service of the piece. These men are stationed within the turret assembly and rotate with it as the gun is traversed. The adoption of a completely automatic tank gun would reduce the size of this crew. The fourth crew member, the driver, is usually located within the hull, as his duties require that he be oriented with respect to the vehicle axis rather than with respect to the armament axis.

The turret and basket assembly is supported by the turret traverse bearing assembly which, in turn, is mounted on a large ring in the top surface of the hull casting (see Fig. 2-11). This turret traverse bearing and its support must be designed to withstand the loads imposed by the firing of the main weapon, by shocks caused by vehicular travel over rough ground or obstacles, by high explosive blasts, and by ballistic impacts against the turret. The sensitivity of sighting, fire control, and stablization equipment requires that clearances in the bearing assembly be limited to an absolute minimum consistent with economical production practices. Both roller bearing and ball bearing type assemblies in a variety of configurations have been used to meet these requirements. The comparative simplicity and minimum space requirements of the single-row ball bearing assembly, however, have led to its adoption as a standard arrangement for traverse bearing assemblies.

Operating space in the turret is limited by several factors. Requirements for maximum ballistic protection lead to thick turret walls, sloping surfaces to avoid re-entrant angles and to deflect attacking projectiles upward or away from the hull roof, and low silhouette. Limitations on maximum vehicle width plus track width requirements for low ground pressure place limitations upon the maximum diameter available for the turret ring which limits the available turret space. Although the turret can extend beyond the diameter of the turret ring above the turret ring level, it cannot

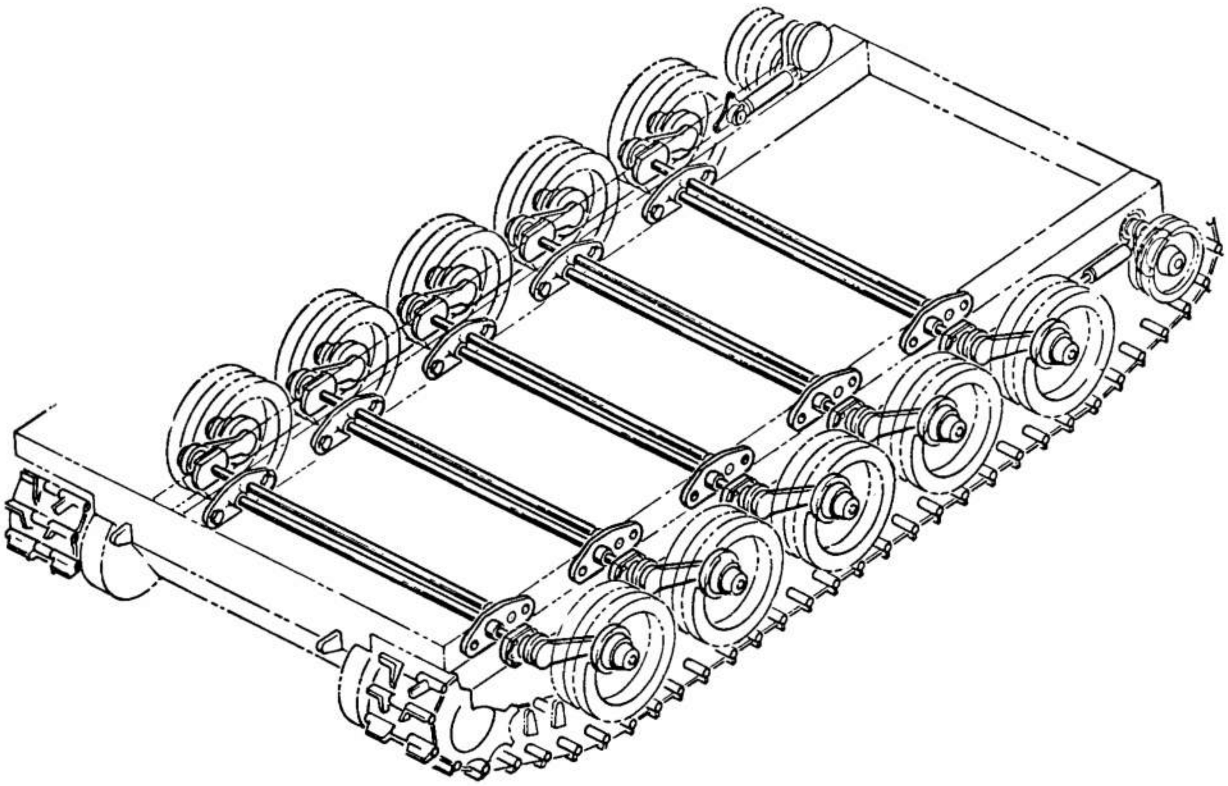

Figure 2-10. Torsion Bar Arrangement in Tracked Vehicle

extend so far as to interfere with engine cooling, the driver's hatch cover, or other hull components.

2-12.4 HULL-WEAPON RELATIONSHIPS

The principal weapon considerations that influence hull design are the types and calibers of the weapons, the weapon mounting locations, the types and supply of ammunition required, and the ventilation requirements imposed by weapon firing. Practically any type of crew-served weapon can be mounted in a hull-type vehicle, and individual weapons can be fired through suitable firing ports. The armament to be mounted on a vehicle is dependent upon the vehicle's mission and upon the requirements and desires of the using agency. This information is available to the designer in the Military Characteristics for the vehicle to be developed.

Weapon characteristics are a prime consideration in the design of the hull of a combat vehicle, particularly those of the primary armament. Obviously, the weapon's size, weight, and trunnion reactions are important considerations; but such factors as maximum angles of elevation, depression, and traverse are equally important, as the hull must permit complete freedom of movement through these desired angles. Furthermore, sufficient clearance must be provided within the vehicle to permit the loading and unloading of the weapon and to accommodate weapon recoil when firing at the limits of its elevation, depression, or traverse. Provisions must also be made in the design for performing routine maintenance and service operations on the weapon, for performing repairs, for replacing major elements such as barrels, recoil mechanisms, or equilibrators, and for removing and replacing the entire weapon assembly when necessary.

Some combat vehicles, e.g., tanks, have their primary (and secondary) weapon mounted in a fully rotatable turret. In such cases, many of the considerations just mentioned become the responsibility of the turret designer; however, the hull designer must coordinate his efforts with the turret designer for a number of reasons. Maximum turret space must be made available

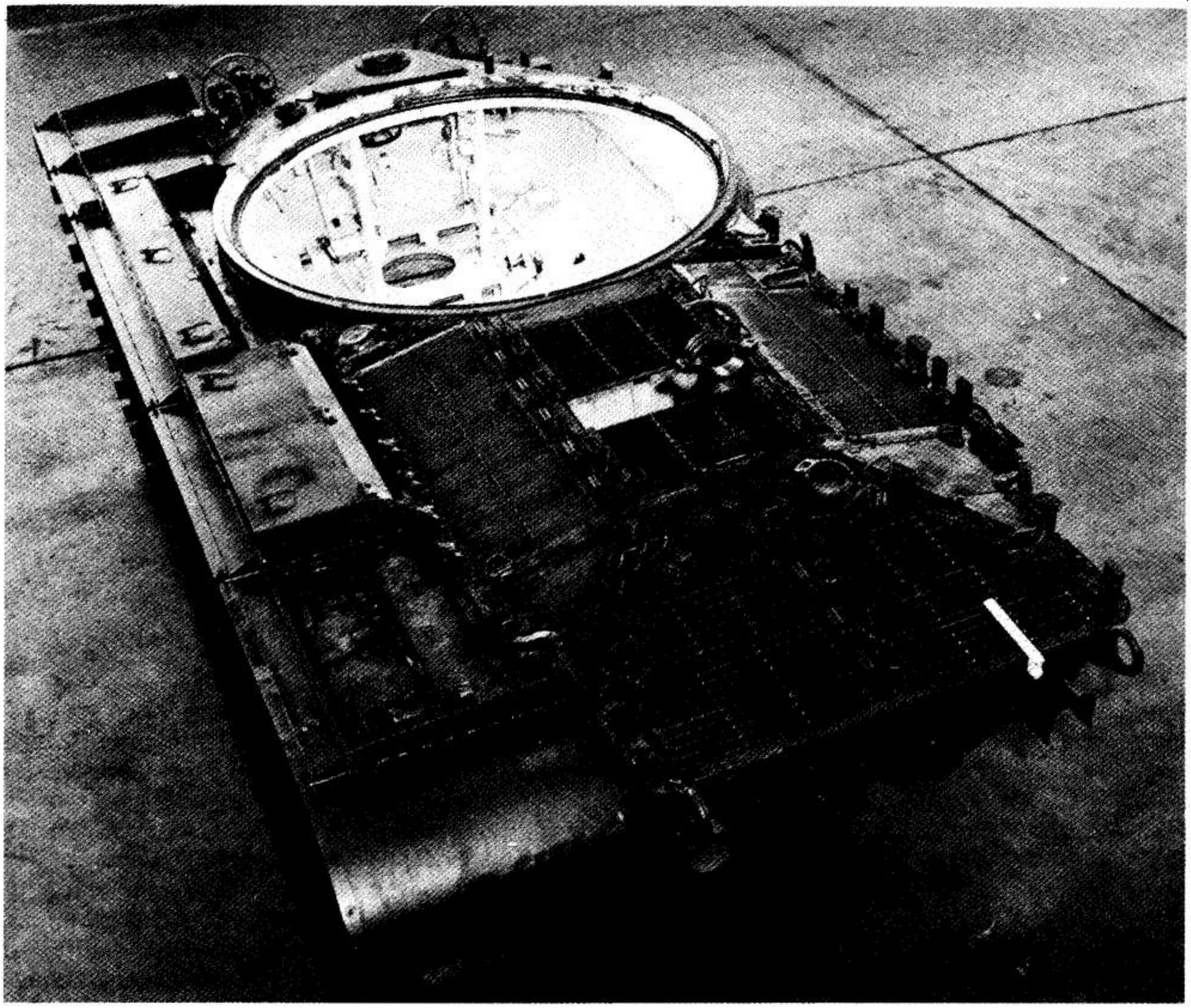

Figure 2-11. View of Medium Tank Hull Showing Turret Race Ring

(the turret designer never has as much as he would like) within the limitations imposed by maximum permissible vehicle width and track width requirements. Consideration of desired weight distribution upon the suspension system influences the location of a heavy turret and weapon system with respect to the hull. The maximum depression angle of the armament influences the hull length and its configuration. Turret overhang influences such considerations as the location of engine air intakes, driver's hatch location, and track and fender height above the hull line.

Current combat tanks are armed with primary weapons ranging in size from 76 mm to 152 mm. A 76 mm gun is commonly used in light tanks, a 90 mm or 105 mm gun in medium tanks, and a 120 mm gun in heavy tanks. The new XM551 Armored Reconnaissance/Airborne Assault Vehicle is a lightweight tank-type vehicle armed with a new 152 mm weapon capable of firing conventional rounds or guided missiles. This weapon is also being installed on some medium tanks. Self-propelled artillery in current use mount weapons of 90 mm, 105 mm, 155 mm, and 8-inch caliber.

Ammunition used in tank guns and artillery are of four general types—fixed, semifixed, separated, and separate-loading ammunition. *Fixed ammunition* is a complete round consisting of a projectile that is crimped securely into a cartridge case which contains the

propelling charge and which has the primer fitted into its base. The complete round is loaded into the weapon as a unit. *Semifixed ammunition* is a round in which the cartridge case is not permanently fixed to the projectile to permit adjustment of the zone charge. The complete round is loaded into the weapon as a unit. *Separated ammunition* is a round that consists of two units—the projectile and the propelling charge. The propelling charge is contained in a primed cartridge case which is sealed at its mouth with a closing plug. When the gun is loaded manually, the projectile is inserted into the breech of the gun and is partially seated by the ramming action of the cartridge case as it is loaded. The closing of the breech completes the seating of the projectile. When the gun is loaded semi-automatically, the projecticle is first rammed home by a mechanical rammer, after which the cartridge case is loaded and the breech is closed. *Separate-loading ammunition* consists of separate components (projectile, propelling charge, and primer) which are loaded separately into the weapon.

Tank guns and self-propelled artillery generally use the fixed and separated types. Fixed ammunition is used in guns up to and including 105 mm, and the larger guns use the separated type because fixed ammunition in calibers larger than 105 mm is too awkward to handle in the cramped confines of a turret. A typical round of 120 mm ammunition consists of a 24-in. long projectile and a 33-in. long cartridge case. The round, assembled as fixed ammunition would be nearly 5 ft long and would weigh approximately 100 lb. Besides being difficult to handle, such rounds would be almost impossible to stow on board the vehicle in any quantity. Separate-loading ammunition is generally used in very large caliber guns and howitzers. Thus, the ammunition requirements of the vehicle armament and the need for on board ammunition stowage pose a real problem to the hull designer. More detailed information on ammunition is given in Refs. 108-111.

The ammunition supply carried by the vehicle depends upon the mission of the vehicle and upon the types of weapons with which the vehicle is armed. The mission requirements influence the number and types of rounds needed, and the weapon types influence the total stowage volume needed for ammunition. These requirements are usually given in the Military Characteristics for the vehicle. Usually enough rounds are stowed in close proximity to each weapon to enable it to complete one fire mission. The remainder of the rounds are stored elsewhere in the hull.

As the weapons are fired, the empty cartridge cases are ejected onto the floor of the vehicle. Despite the use of bore evacuators in large guns, rapid sustained fire with the tank buttoned up often fills the fighting compartment with noxious fumes. This contaimination is due to the vacuum created at the breech by the ejection of tightly fitting cartridge cases and to the fumes which emanate from ejected cartridge cases. Under these conditions, forced ventilation must be employed to prevent these fumes from impairing the efficiency of the crew. In turreted vehicles, a ventilating blower is provided in the turret with an armored inlet in the turret roof. This allows the air flow to be directed easily to the breech area of the weapon to force the residual propellant fumes out through the gun muzzle; and in addition, locating the air inlet in the relatively thin roof does not present armor protection problems that are as expensive to solve as would a hull location.

2-12.5 HULL-POWER PLANT RELATIONSHIPS

Strictly speaking, the power plant comprises the engine assembly including its cooling system and all necessary accessories for its operation such as starters, electrical system, lubrication system, etc. In tank-type vehicles, however, the transmission system is so closely integrated with the power plant that the two systems form a single power package. This arrangement saves space in the vehicle and simplifies its installation and removal. In this discussion, the term "power plant" refers to the entire power package.

The vehicle power plant is a major subsystem that requires consideration, not only because of its functional importance but also because of its weight, bulk, operational requirements, and because of the reactive forces it transmits to the hull. Its weight and bulk are major contributing factors to the size and weight of the vehicle itself. Being vulnerable to damage by ballistic and explosive projectiles, it must be protected by armor; and being inherently large, this accounts for an appreciable amount of armor. It has been estimated that, in a medium tank, every inch of power package height may require

as much as 430 lb of armor for each inch of hull silhouette; every inch of power package width may require 180 lb of armor; and every inch of length, 90 lb of armor.

Another factor for consideration is that of power plant location within the vehicle. Being a heavy item, especially when also credited with the weight of its associated hull armor, it is best located as low in the hull as other considerations permit. Generally, this leads to a lower center of gravity for the vehicle, a lower hull silhouette, and reduces the possibility of the hull interfering with the main armament at maximum weapon depression and for a given trunnion height. Longitudinally, the power plant is usually located toward one end of the vehicle and oriented with the engine inboard of the transmission, i.e., with the engine toward the center of the vehicle, and the transmission toward the end. This orientation places the transmission output shaft more conveniently with respect to the track driving sprockets.

The choice between rear drive and front drive is largely a matter of power plant vulnerability and weight distribution. Combat vehicles—such as tanks, combat reconnaissance vehicles, assault vehicles—and various close support vehicles—such as medium and heavy tank recovery vehicles and tracked combat engineer vehicles—are required to operate at the forward edge of the battle area and even behind the enemy lines. As such, they are most likely to encounter frontal attack by high-powered direct fire weapons. This situation dictates that their frontal areas be protected with extremely heavy armor. Locating the power package in the front of these vehicles would increase the probability of it being damaged and would greatly increase the weight penalty required to protect it with adequate armor. Thus, these types of vehicles generally have their power plants located at the rear.

On the other hand, vehicles of the self-propelled artillery type, by the nature of their missions, are less likely to encounter frontal attack by direct fire weapons and can, therefore, dispense with heavy armor in this area. Furthermore, the weight, recoil characteristics, and operational requirements of the heavier guns with which they are armed usually make it more desirable to mount these weapons at the vehicle rear and locate the power plant at the front. This makes maximum space available for service of the piece and for convenient ammunition stowage. Locating the power plant at the front also tends to counterbalance the weight of the weapon and serves to distribute the weight more advantageously on the suspension system.

Armored personnel carriers, tracked cargo carriers, special-purpose amphibious vehicles, and vehicles of this general category generally require an unobstructed cargo hold at their rear. Locating the power plant in the front provides this space at the rear and also helps equalize the load on the suspension system. Control of the power plant and of the vehicle, since a tracked vehicle is steered through the transmission system, is simplified somewhat when the power package is located up front because the driver's station is generally located up front. This makes for shorter and simpler control linkages.

The power plant requires large quantities of air, for the engine to consume and for the cooling of the engine and transmission fluid. This air is generally brought in through vents and grilles located in the top deck of the hull directly over the engine and transmission compartment. The engine compartment is separated from the other compartments of the vehicle by welded bulkheads. Access doors are provided in the bulkheads where necessary for access to service and maintenance points on the engine and transmission. On some vehicles, such as the medium tanks, the power plant compartment is permitted to flood during deep-fording operations. In these vehicles, the access doors in the bulkheads must be watertight to prevent flooding of the crew compartment. The bulkhead which isolates the power plant from the remainder of the vehicle serves other purposes. It isolates the crew from power plant fumes, heat, and, to some extent, noise. It also serves as a firewall in case of an engine fire. Seals must be provided around all linkages, electrical lines, hydraulic lines, and control lines passing through the bulkhead. This is particularily important in vehicles designed to operate in CBR contaminated areas.

A prime consideration of power plant-hull relationships is adequate provisions for power plant servicing, inspection, maintenance, repair, and replacement. Toward this end, access doors and removable panels are provided in the hull where necessary. Access doors and openings should be large enough to accommodate the

operation for which they are intended. If intended for the replacement of a defective part, they must be sufficiently large to accommodate the part and any tools, slings, fixtures, hands, and arms that may be necessary to complete the task. Ref. 32 gives excellent guidance on the dimensional requirements of various maintenance tasks. Provisions should be made in the hull design for the removal of the complete power package. It should be remembered that the power package is extremely heavy and awkward to handle. Therefore, hull openings for power package removal should be ample in size and positioned to permit vertical removal without any need for lateral jockeying of the heavy assembly.

In addition to the factors mentioned, the hull: (1) provides secure mounting for the power plant, (2) must be capable of absorbing the torques, vibrations, and reaction loads, and, (3) maintains correct power plant alignment with the balance of the drive train. It also provides adequate air space around the power plant to insure adequate cooling with minimum air flow restrictions. The "breathing room" around the power package should be made as large as is practical and consistent with the overall vehicle space requirements.

2-13 ARMORED VEHICLE HULLS[69,112]

2-13.1 BASIC DESIGN CONSIDERATIONS

2-13.1.1 General

Basic considerations in the design of armored hull structures are the degree of protection required, the volume required, and the maximum allowable weight. The weight of the armor must be no greater than the difference between the maximum allowable weight of the vehicle and the weight of the remainder of its components. This limitation is not merely a residual factor, but one which has been derived from necessary compromises resulting from the interrelation of firepower, mobility, and protection. The problem, then, involves the distribution of the permissible weight and type of armor which gives the maximum protection.

2-13.1.2 Probability of Projectile Impacts

The probability of projectile impacts is an important factor in determining thickness and obliquity of armor. This probability is controlled by the following basic factors:

a. The size of the area in relation to the total projected area of the vehicle in the plane normal to the line of expected attack

b. The location of the area in relation to the probable aiming point

c. The probability of exposure of the area to enemy fire; i.e., the amount by which the area is reduced when protection and concealment are provided by terrain

Study of these factors over a period of time has established a design axiom that the heaviest attack will be against the front wall of a tank and then against the sides, top, floor, and rear in decreasing order.

It should be noted that each of the above factors is in itself dependent upon a probability such as "expected attack", "probable aim point", and the "probability of exposure". The probabilities associated with these factors are usually determined through empirical means by analyzing combat records. Naturally, since there are a variety of vehicles, missions, combat situations, classes of enemy, etc., a great deal of judgment is needed in making an armor requirements analysis. Ideally, the armor distribution for each vehicle type should include the broad spectrum of its mission and the types and probability of use of the various weapons which may be encountered.

2-13.1.3 Equalization of Protection

Another factor to be considered in determining the thickness and obliquity of armor is equalization of protection. This means that if the armor of the vehicle has been properly designed, each portion of the vehicle has an equal probability of surviving an attack under the large majority of expected combat encounters. In order to obtain equalization of protection, the positional frequency distribution, with respect to the vehicle, of the weapons expected to be encountered is required in addition to the data discussed previously. For instance, the positional frequency distribution could be such that the vehicle may receive fire from the flanks in 75 percent of the encounters and from the rear, front, and top in 10, 10, and 5 percent of the encounters, respectively. Thus, the equalization of protection of the vehicle

would require the heaviest armor on the sides, lesser armor on the front and rear and the least armor protection on the top.

A report prepared by the British, based on approximately 1600 tank casualties in combat during World War II, summarizes information regarding the probability of projectile hits on various parts of the tank. About 64 percent of the hits were on the hull and suspension and 35 percent were on the turret. Approximately 40 percent of all of these hits were on the frontal areas. Although tank hulls received almost twice as many hits as turrets, the proportion of hits on turrets was about twice as great as would normally be expected for the area exposed. This condition is a result of the turret being exposed first above a hill or any other type of concealment.

Fig. 2-12 illustrates the equalization of protection for a World War II tank. This figure illustrates the need to consider weapon caliber in addition to frequency of hit encounters. In the theoretical combat situation illustrated, maximum expected attacks against the main portions of the tank are shown by solid arrows. If this tank is to defeat the attack, each section must incorporate equalized thicknesses and obliquities sufficient to meet its design requirements.

When attacks from directions other than direct front, flank, or rear are considered, additional means must be employed to maintain equalization of protection. For example, if the attack were from somewhere between direct front and direct flank, the following suggestions might be used to establish the armor design:

a. The limiting angle of attack from the front for the 90 mm projectile would be the angle at which the projectile, with a striking velocity equal to that used in direct frontal attack, would just be defeated by the basic armor on the side.

b. Likewise, the limiting angle of flank attack for the 75 mm projectile would be that angle at which the projectile, with the same striking velocity as that of the direct flank attack, would just be defeated by the basic rear armor.

c. Therefore, in the design of these areas or components subject to attacks somewhere between direct front and direct flank, the 90 mm projectile would be considered for angles of attack for that projectile.

d. In the case of attack somewhere between direct rear and direct flank, the 75 mm projectile would be considered for angles of attack up to its limiting angle of flank attack. The armor thickness and obliquities would be established to defeat projectiles with striking velocities equal to those established for direct attacks.

2-13.1.4 Re-entrant Angles

A re-entrant angle is an angle formed by two surfaces of a vehicle such that a projectile striking either surface may be ricocheted against the other surface. This condition is undersirable, because projectiles may be deflected to surfaces affording inadequate protection, or they may be deflected so as to strike the second surface at a lower angle of obliquity, thus possibly defeating the armor. Re-entrant angles should be avoided; however, if they are required, the sloping surfaces should be arranged so that adequate protection is afforded for deflected projectiles.

2-13.1.5 Armor Surface Conditions

Irregularities in a surface, either inside or outside of the vehicle, tend to create weaknesses in the structure and should be avoided. A flat, smooth wall of constant thickness offers the best resistance to attack because the armor can absorb the shock impact uniformly over a larger area. Any irregularity—whether it be a reinforcing brace, a protective bead, a sudden change in thickness, or a sudden change in obliquity such as a welded joint—tends to offer a stress concentration and restricts uniform deformation required for maximum energy absorption.

2-13.1.6 Armor Basis

Armor basis is a means of evaluating the amount of armor protection afforded by various armor walls against armor-piercing projectiles. Armor basis is defined as the thickness of homogeneous armor required at 0° obliquity to give ballistic protection equal to a specific thickness of armor at a given angle of obliquity. The angle of obliquity is defined as the angle between the projectile trajectory and the normal to the surface of the armor plate. This relationship is illustrated in Fig. 2-13. The armor basis of a given plate is accurately determined

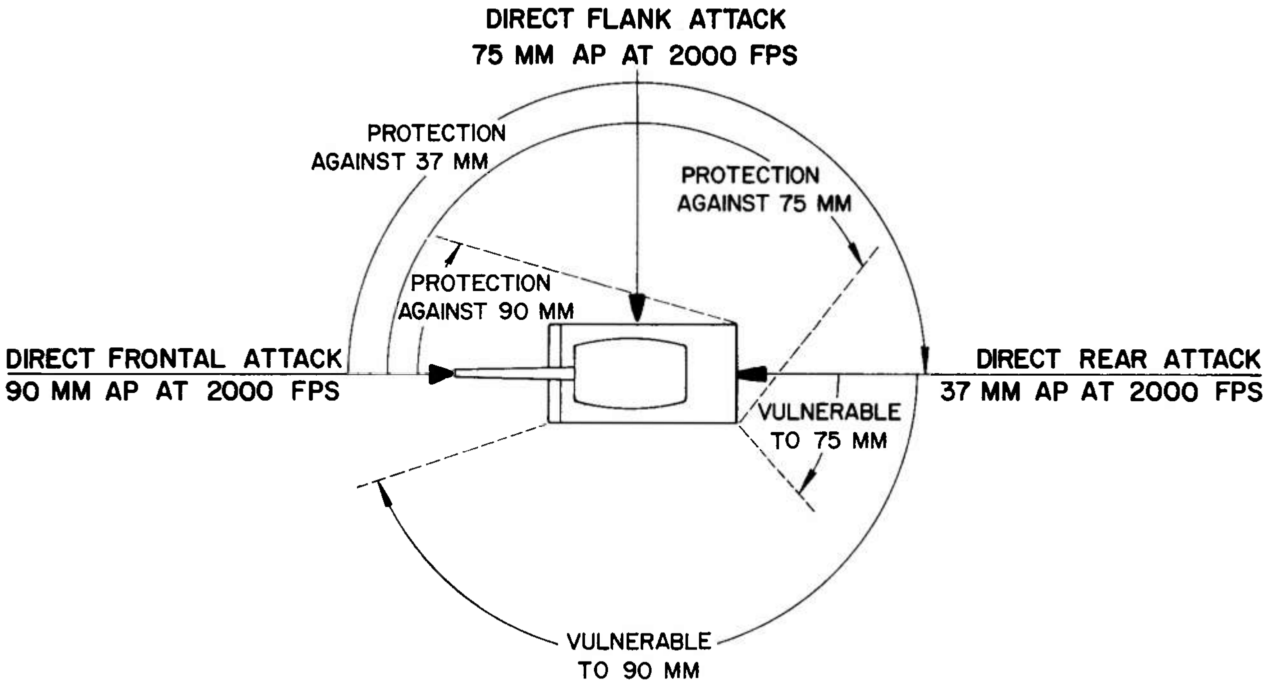

Figure 2-12. Equalization of Protection

only under conditions of equivalent ballistic protection against a specific type and caliber of projectile.

Currently, armor basis is presented at basic angles more nearly approximating the desired thickness and angle of obliquity. For example, the requirement might specify a frontal armor basis of 4.0 in. at 60° obliquity for a certain range (or impact velocity) and projectile. By referring to appropriate thickness-obliquity charts[69] the designer can select thickness and obliquity combinations to effect weight and space compromises.

2-13.1.7 Ballistic Limit

2-13.1.7.1 Army Ballistic Limit

The Army Ballistic Limit is the critical or limiting velocity according to Army criterion at which a specified projectile will be borderline in penetrating the armor being impacted. In this ballistic test, a complete penetration occurs whenever (1) a projectile or fragment has penetrated the armor sufficiently to permit at least a pinhole of light to pass through a hole or crack developed in the armor, or (2) the front of the fragment or nose of the projectile can be seen from the rear of the armor. A partial penetration occurs when lesser damage to the armor occurs.

2-13.1.7.2 Navy Ballistic Limit

The Navy Ballistic Limit is the critical or limiting velocity according to Navy criterion at which the specified projectile will be borderline in penetrating the armor being impacted. In ballistic tests to determine this limit, a complete penetration is considered to occur whenever the entire projectile, or essentially the entire

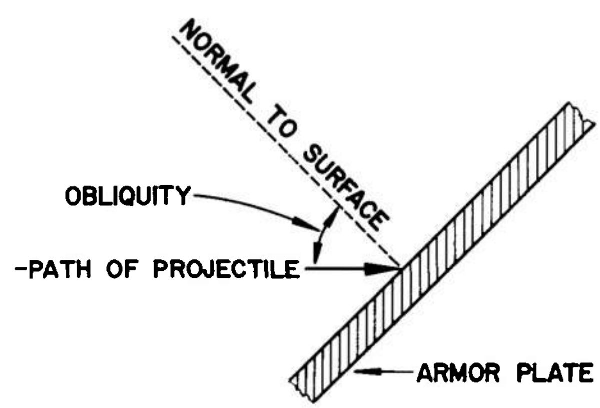

Figure 2-13. Illustration of Obliquity

projectile, passes completely through the armor. All other penetrations are classified as partial. No witness plates are employed in these tests.

2-13.1.7.3 Protection Ballistic Limit

The Protection Ballistic Limit is the critical or limiting velocity at which the specified projectile will be borderline in penetrating the armor being impacted, according to the *protection criterion.* In this case, a complete penetration is considered to occur whenever a fragment or fragments from either the impacting projectile or the armor are caused to be thrown from the back of the armor with sufficient remaining energy to pierce a sheet of 0.020-in.-thick 2024-T3 aluminum alloy placed parallel to and 6 in. behind the target. A fragment with this amount of energy is normally expected to produce lethal damage if it strikes a vulnerable area. Any fair impact which rebounds from the target, remains embedded in the target, or passes through the target, but with insufficient energy to pierce the 0.020-in.-thick aluminum alloy witness plate, is termed a partial penetration.

2-13.1.7.4 V_{50} Ballistic Limit

The V_{50} Ballistic Limit is a critical velocity of a fragment or a projectile at which 50 percent complete penetrations and 50 percent partial penetrations of the armor target can be expected on a limited statistical test. Above this limit, complete penetrations of the armor will generally predominate; below this limit, partial penetrations will predominate.

2-13.1.7.5 Zone of Mixed Results

The inherent variables within a material and the variables in any ballistic test such as a slight difference in weights of projectiles and the orientation of projectiles at the instant of impact, introduces a probable *zone of mixed results* into the test. As the name implies this zone of mixed results may contain one or more impacts that completely penetrated the material under test at velocities below those of other impacts that failed to effect complete penetration. Velocities in this zone of mixed results can vary up to several hundred feet per second depending upon projectile reaction and the mechanism of penetration.

2-13.2 TYPES OF ANTITANK PROJECTILES[108,109]

2-13.2.1 General Types

Antitank projectiles may be divided into two general catagories—kinetic energy and chemical energy projectiles. The effect of kinetic energy projectiles, including various armor-piercing types, depends on projectile mass and impact velocity. The effect of chemical energy projectiles, including various high explosive types, depends primarily on explosive action and relatively little on the striking velocity.

2-13.2.2 Kinetic Energy Projectiles

2-13.2.2.1 Ball Projectiles

Ball projectiles usually employ a soft metal core and a jacket of clad steel or gilding metal. Complete penetration of any normal thickness of tank armor is practically impossible with this type of projectile. Any damage which might result is confined mainly to bullet splash or to keying, in which the bullet becomes wedged between the turret and the hull.

2-13.2.2.2 Armor-piercing (AP) Projectiles

An armor-piercing (AP) projectile has a solid, high-carbon-alloy steel body that is heat-treated to give it a tough core, to withstand transverse stresses when the projectile strikes armor at an angle, and a very hard nose for armor penetration.

The armor-piercing capped (APC) projectile was designed especially for penetrating face-hardened armor. It employs a main body of alloy steel, similar to that of armor- piercing projectiles, and has a forged alloy steel cap, heat-treated to have a hard face and a relatively soft core, attached to the projectile nose. On impact, the hard face of the cap destroys the hardened surface of the armor plate, while the softer core of the cap protects the nose from breaking up on contact with the armor. A steel or aluminum windshield is often placed over most armor-piercing projectiles to improve exterior ballistics, and a tracer may be present in the base. Some APC projectiles also contain an explosive charge and fuze in their bases.

2-13.2.2.3 High Velocity Armor-piercing (HVAP) Projectiles

The hypervelocity or high velocity (see Glossary) armor-piercing (HVAP) projectile is

lightweight and consists of an extremely hard core of tungsten carbide, a steel base which contains a tracer element, an aluminum outer body and nose plug, and an aluminum windshield. Projectiles of this type are very effective in penetrating appreciable thicknesses of armor due to their extremely high velocity, high core density, and a core of relatively small cross section.

A variation of this type of projectile is the hypervelocity armor-piercing discarding sabot (HVAPDS) projectile. The discarding sabot is a counterpart of the lightweight outer body of the conventional HVAP but of an increased outside diameter to fit a large caliber gun. After leaving the muzzle, the sabot falls away from the core to eliminate the high drag that is inherent to large diameter projectiles. This concept allows the firing of long, slender tungsten carbide projectiles from large caliber guns to attain high velocities.

2-13.2.3 Chemical Energy Projectiles

2-13.2.3.1 High Explosive (HE) Projectiles

The high explosive (HE) projectile is designed for use in antipersonnel roles. This projectile is made of die forged steel with a central cavity that contains a large bursting charge of high explosive. The effect on armor is the result of both fragments and blast.

2-13.2.3.2 Squash Head, or Plastic Charge, Projectiles

The squash head, or plastic charge, projectile contains a plastic explosive. It is made with thin steel walls and is fitted with a base-detonating fuze. Upon striking an armored surface, the projectile ruptures and allows the plastic to mushroom on the armor before it detonates. The detonation produced shock stresses are transmitted through the plate and are reflected as tensile stresses at the rear surface of the armor. As these stresses exceed the material capabilities, large, jagged spall fragements are ejected a high velocity from the back of the armor plate. The armor is thus defeated with little noticable damage to the exterior.

2-13.2.3.3 High Explosive Antitank (HEAT) Projectiles

The functional concept of the high explosive antitank (HEAT) projectile is based upon the shaped charge principle. Effectiveness of this projectile results from the formation of a supervelocity, extremely dense jet of molten metal capable of penetrating a considerable thickness of armor. The unique construction of the shaped charge is responsible for the jet. Basically, the shaped charge consists of an explosive filler with a metal-lined, conical cavity at the front and a detonator at the base. This construction is shown in Fig. 2-14(A). Means are also provided in front of the charge to maintain a stand-off distance from the armor of several inches at the time of impact. When the filler, explodes, lines of force, acting perpendicular to the wall of the liner, cause the wall to collapse and the resultant particles to intersect along the perpendicular altitude of the cone. Fig. 2-14(B) illustrates this focusing of liner particles. The particles, traveling at velocities up to 30,000 fps, drive or force a hole through the armor. Penetration is not the result of a "fusing" or burning action, but rather a function of the velocity of the liner particles. Hence, heat derived from the acronym HEAT, is a misleading term. Damage, or kill, is caused by molten particles of armor plate and the liner jet entering the tank at high velocities and striking vulnerable components or personnel.

To maintain pace with shaped charge development, the armor designer has investigated the use of various armor arrangements, such as spaced armor and barbed spikes. These armor arrangements tend to break up the cone so that the jet will not form effectively. Special purpose materials, such as glass, plastics, and agglomerates, have also been tried. Whenever any of the tested methods or devices appeared promising, weight and space limitations imposed by correct basic tank design requirements prevented their application. However, still other materials and methods of defeating the shaped charge are being investigated and may satisfactorily solve the problem in the future.

2-13.3 MODES OF ARMOR DAMAGE

2-13.3.1 Penetrations

2-13.3.1.1 Factors Affecting Penetrations

In addition to the metallurgical characteristics of the armor plate and the design of the

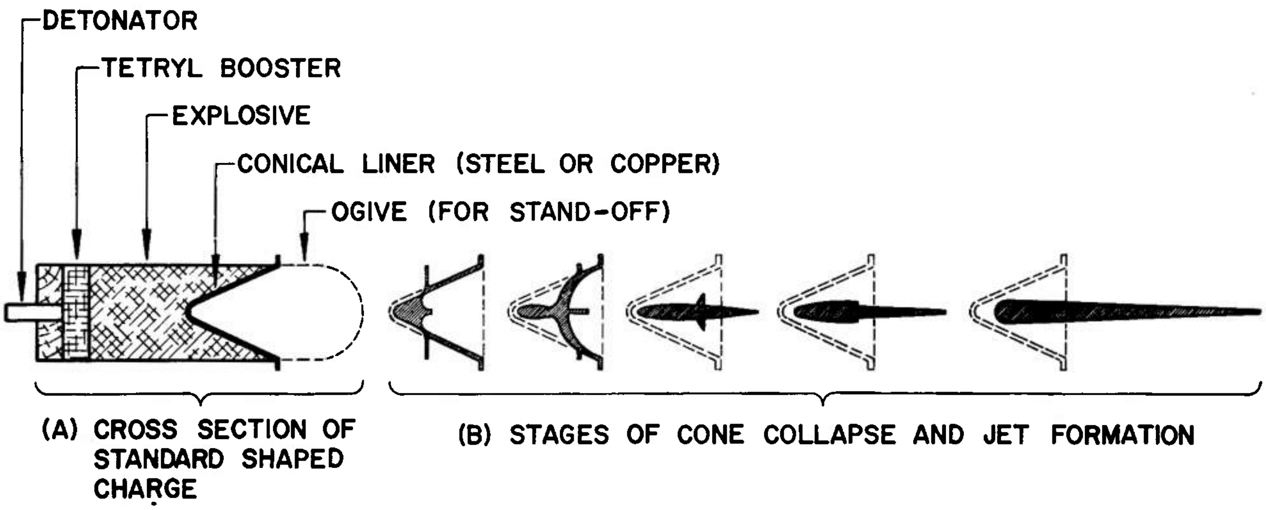

Figure 2-14. Shaped Charge Principle

attacking projectile, four factors determine the depth of penetration of kinetic energy projectiles:

a. Striking velocity
b. Diameter of projectile
c. Thickness of armor
d. Angle of obliquity

Essentially, the striking velocity of a given projectile is a function of the range and muzzle velocity. The depth of penetration of a projectile depends, to a great extent, on this striking velocity.

One of the major considerations in penetration is the relationship of armor thickness and projectile diameter. The relationship of armor thickness and projectile diameter is often expressed as t/d, where t is the plate thickness and d is the projectile diameter (both t and d are expressed in inches). If t/d is greater than one, the projectile is said to be undermatching in relation to the armor thickness; if t/d is equal to one, the projectile is considered matching; and if t/d is less than one, the projectile is overmatching. The performance of projectiles of difference calibers are somewhat comparable when the t/d ratio remains constant. For undermatching projectiles, other factors being equal, the performance of the round is directly proportional to the diameter.

Obliquity increases resistance to penetration over that at normal impact. The reason for this is twofold; (1) a greater thickness of armor is presented to the path of the projectile; and (2) the mechanism of penetration is different. The armor may reject or deflect the projectile, or at least offer enough resistance to the missile to decrease the residual velocity below that required for penetration.

In an actual attack, obliquity is influenced by the following factors:

a. The relative location of armor plate and gun seldom permits normal attack even on a vertical plane.

b. The armor plate is usually sloped from the vertical.

c. The trajectory of the projectile is always a curve.

d. The axis of the projectile may deviate from true line of flight (yaw).

e. The pitching and rolling of a moving vehicle, or the slope of the ground under a parked vehicle, influence the effective slope of the armor plate.

The effects of trajectory and yaw are small for well-designed projectiles at short ranges and are usually neglected. Similarly, the factors that influence the effective slope of the armor plate are continually variable and are not considered in armor design.

Since compound conditions in an actual attack result in angular fire at a sloped plate, the true angle of obliquity becomes greater than either the angle of attack or the slope of the plate. This condition is illustrated in Fig. 2-15, where the cosine of the true angle of obliquity is shown to be equal to the product of the cosine of the angle of attack and the cosine of the angle

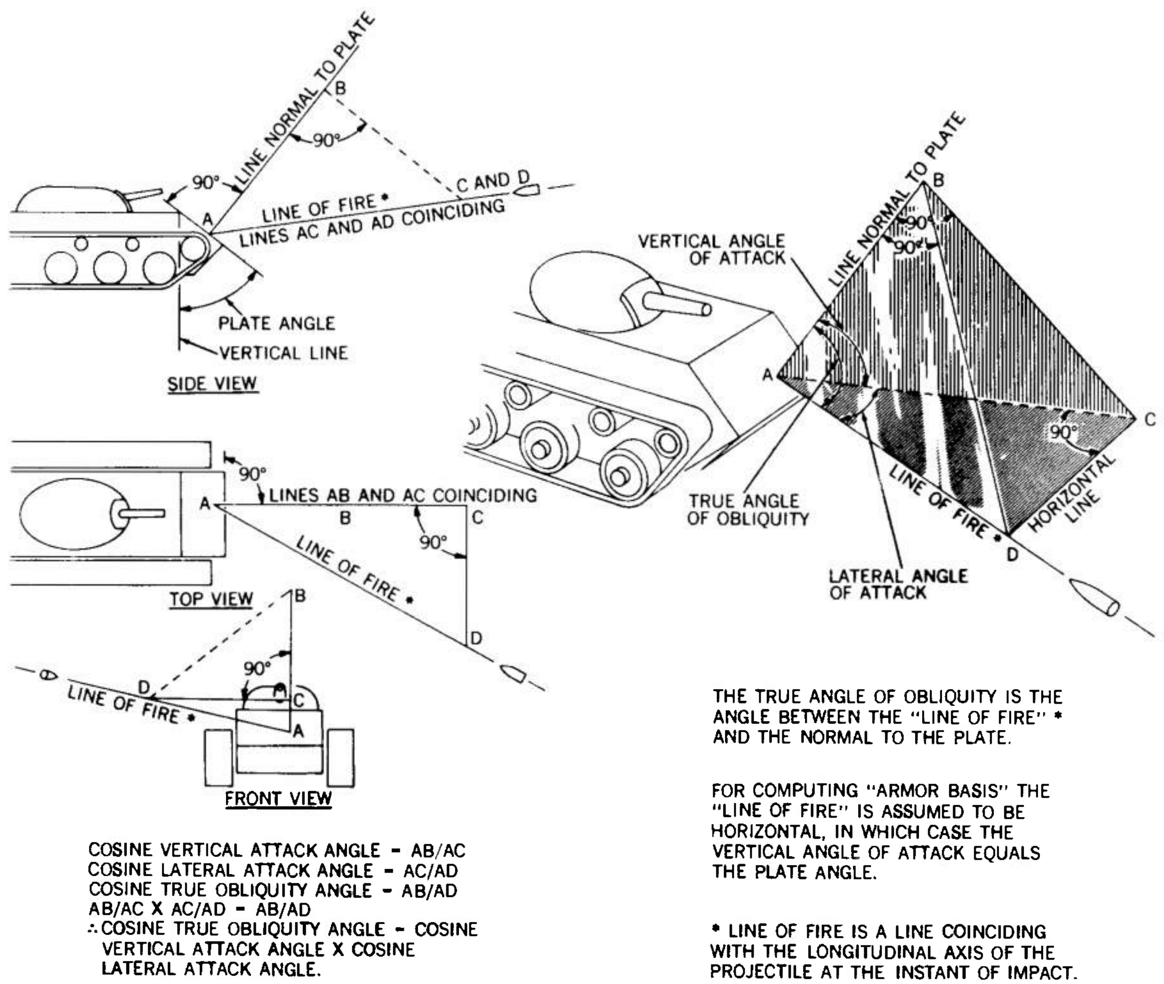

Figure 2-15. Determination of the True Angle of Obliquity [112]

of the plate. In Fig. 2-16, the angle of obliquity is plotted as a function of the angle of attack and the inclination of the plate.

2-13.3.1.2 Mechanism of Penetration

With no obliquity and a projectile that is harder than the plate, the displacement of metal necessary to permit penetration occurs in two ways: (1) by a flowing of the metal, and (2) by the thrusting aside or the driving out of pieces of the plate. Fig. 2-17(A) shows a projectile that has penetrated to the depth of its ogive. The metal under the plate surface offers great resistance to the deformation or flow of the metal toward the rear of the plate, and so the metal is pushed to the side and toward the face of the plate forming a bulge. As the projectile penetrates farther, the bulge increases in size until the face of the bulge cracks to form face petals (petalling) as shown in Fig. 2-17(B). This mechanism of penetration is typical of undermatching projectiles at normal velocities (less than 5,000 fps). Cratering occurs when the petals break away from the plate as a result of inferior toughness characteristics.

Occasionally, punching results instead of petalling. Fig. 2-17(C) shows that punching occurs when a metal plug is partially or completely extruded from the rear surface of the armor plate as the result of projectile impact. Punching is typical of overmatching projectiles.

Spalling, the displacement of sections of metal inside the surface of the armor, is shown in Fig. 2-17(D). It is usually caused by poor quality armor steel or by a shock inducing projectile, and is characterized by an exit hole larger than the entrance hole. Spalling may occur without complete penetrations and may

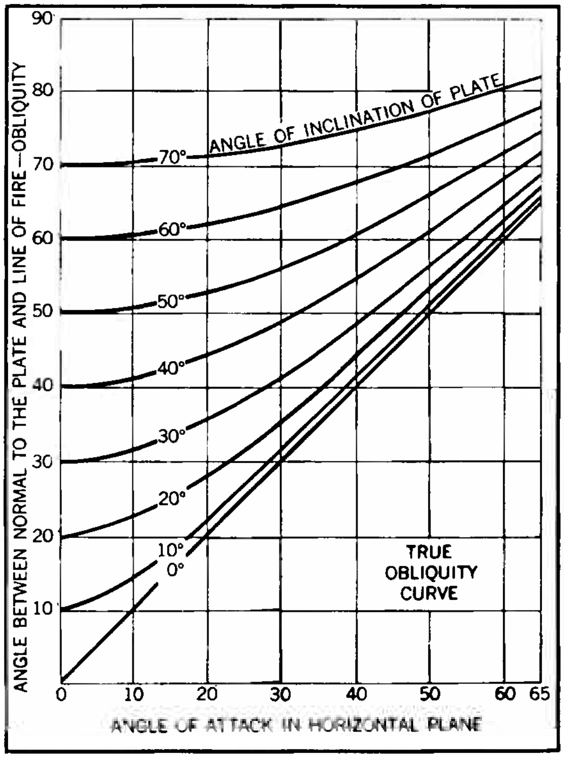

Figure 2-16. True Obliquity Curve [112]

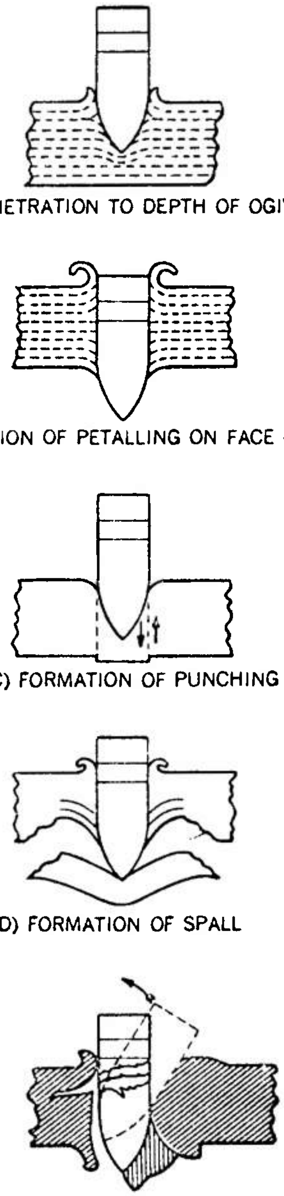

Figure 2-17. Mechanism of Penetration

result in the defeat of a vehicle which ordinarily is invulnerable to the projectile causing the spall.

If a projectile that is harder than the plate impacts the plate at an obliquity, the projectile undergoes bending stesses which tend to break it transversely. This breaking-up is more frequent at oblique impact than at normal impact; hence the results are more variable. High obliquity, of course, tends to cause the projectile to glance off the plate and leave a long shallow impression. At lower obliquities, the projectile tends to dig in, causing petalling on one side of the hole and, perhaps, penetration in the form of a punching through the thickness of the plate. If punching does result, the projectile tends to right itself in the hole and may deform or break. Fig. 2-17(E) demonstrates the mechanism of this type of penetration at obliquity.

2-13.3.2 Shock

Resistance to shock is that property of armor which enables it to absorb, without cracking or general rupture, energy from the impact of kinetic energy projectiles or from the explosion of high explosive projectiles. Because of the very high striking velocities of many projectiles and the great force with which explosions occur, the energy absorption must take place in a very short period of time. A proper relationship of material toughness and hardness must be established to obtain optimum resistance to shock.

Another effect of shock is armor vibration. Although armor vibrations resulting from shock do not damage the armor, equipment mounted in the vehicle near the shock area may be shaken

loose from the armor wall. To minimize shock effects, the designer attempts, within limits, to observe the following armor design criteria.

a. Take advantage of isolated areas for fragile equipment.

b. Design shock mounts to minimize transmittal of energy.

c. Maintain sufficient space between mounted equipment and armor wall to prevent damage from impact bulges.

d. Avoid bolting equipment or mountings directly to an exposed armor wall.

f. Attach electric cables with clips that will be retained on the cable if displaced.

2-13.3.3 Blast

In an explosion, the sudden release of large amounts of energy results in a high pressure wave front. Unless a direct hit occurs, the effects of blast are generally absorbed by the hull structure of current armored vehicles, and only shock damage is experienced in the form of a general shaking loose of internally stored equipment and possibly the springing of some joints. A near miss can cause local buckling of the sides, particularly in the case of the lighter vehicles. Land mines develop a considerable blast which requires the hull floor to be reinforced accordingly. Since land mine detonation will usually occur beneath the front section of an armored vehicle, the front section of the underplate is made heavier than the rear section.

2-13.3.4 Bullet Splash

Bullet splash is the dispersion of finely divided or melted metal produced upon impact of a projectile with armor plate or other hard objects. These fragments travel at extremely high speeds and are capable of injuring personnel or damaging equipment. In general, bullet splash travels along a plane tangent to the armor at the point of impact. Consequently, the bullet splash from impacts on a convex surface will travel away from the surface of the armor; while splash from impacts against a flat or a concave surface will travel along the surface until it becomes convex, or until the bullet splash is deflected by an irregularity in the surface (Fig. 2-18). Oblique impacts produce splash concentrated toward the direction of original flight; although even under these conditions the splash will normally occur 360° around the impact area.

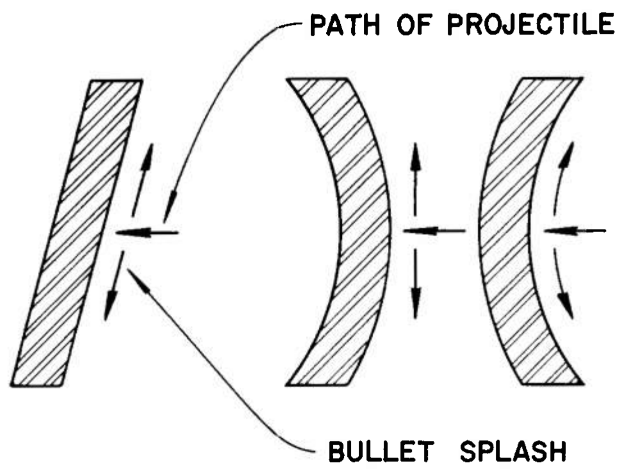

Figure 2-18. Characteristics of Bullet Splash on Various Surfaces

Because bullet splash behaves as a high velocity fluid, it can be turned in several directions and still cause damage. Three right-angle turns are considered the minimum number necessary to expend the harmful energy of bullet splash. Since bullet splash can pass through relatively small openings, careful attention must be given to its control at hatch covers, air vents, vision ports, gun shields, or anywhere else an opening occurs in the armor structure. Bullet splash is controlled in combat vehicles by means of baffles or traps. These deflect the splash and absorb its kinetic energy. In some cases, the splash is turned back along its original course by means of specially designed deflecting surfaces. An illustration of a splash trap is shown in Figs. 2-19 and 3-5.

2-13.3.5 Immobilization

Immobilization, as it relates to armor damage, refers to the rending of any functionally movable part of a structure immovable by means of ballistic attack. This type of armor damage is particularily associated with undermatching projectiles, but is also possible with matching and overmatching projectiles. Three most common types of immobilization are burring, keying, and deformation. *Burring* is the cracking or tearing of the surface or edge of an armor plate as a result of projectile impacts. When two such surfaces are close together, the raised edges

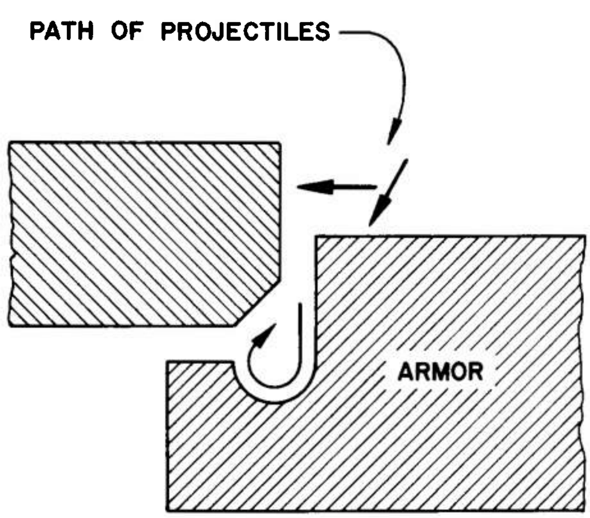

Figure 2-19. Typical Splash Trap

may prevent or impede motion between them. *Keying* occurs when a projectile or fragment becomes wedged between two surfaces, or when a projectile penetrates one surface and partially penetrates the other surface to lock or pin the two movable parts together. *Deformation* is the forcing of a component out of shape, or the swelling of the metal as a result of projectile impacts. This jams the moving surfaces together and prevents normal operation.

In designing exterior moving structures, consideration should be given to protecting or rendering them insensitive to these types of immobilization. Toward this end, the use of projectile deflecting strips and generous clearances between moving parts are two techniques often employed. An example is the application of deflector strips to the base of turret rings. Large clearances between moving parts offer protection against immobilization, but they increase the need and importance of protection from bullet splash.

2-13.3.6 Effects of Nuclear Weapons on Armor

The detonation of nuclear weapons produces large blast effects along with thermal and nuclear radiations. The blast effects are similar to those of any other high explosive weapon (see par. 2-13.3.3). Considerations of thermal radiation require the use of fire-resistant materials for external components and may require special thermal insulation on the inside of the hull to prevent the ignition of combustible materials and ammunition. Initial nuclear radiation may produce induced radiation in the hull. This is discussed briefly in par. 2-6.3.3.3. A comprehensive treatment of nuclear weapon effects on armored vehicles is beyond the scope of this handbook. For a general understanding of the magnitudes involved, Table 2-6 gives the pressure and radiation from various yield weapons at varying distances from the detonation.

2-13.4 ARMOR MATERIALS[69]

2-13.4.1 Steel Armor

Steel is still the primary material for armored vehicles because of its strength, toughness, hardness, workability, joinability, and relatively reasonable cost. Steel armor is made in three principal types: (1) cast homogeneous armor, (2) wrought homogeneous armor, and (3) face-hardened armor. The selection of the type, composition, and metallurgical structure of the armor for a particular vehicle depends upon the following factors:

a. The types of attack the vehicle is required to survive

b. The formability limits of the various types and thicknesses or armor being considered

c. Restrictions imposed by the vehicle design

d. Weldability of the materials under consideration

e. Suitability of materials to flame cutting operations

f. Machinability of the materials

g. Hardness, toughness, and soundness requirements

h. Production capacities of steel mills for particular armor materials

i. The effect on stock pile of strategic alloying elements

2-13.4.1.1 Cast Homogeneous Armor

Cast homogeneous armor is a special alloy steel of high hardenability that has been poured into a mold and later heat-treated to obtain a desired metallurgical structure that is essentially uniform throughout the casting. Cast armor is widely used in the manufacture of medium and heavy tanks. Casting and heat-treating techniques have progressed to a point which makes it practical to cast almost the entire hull

TABLE 2-6 NUCLEAR WEAPON EFFECTS FOR AIR BURSTS WITH MAXIMIZED RANGES[76]

Distances from Ground Zero	*Explosion Yield*				
	1 KT	10 KT	100 KT	1 MT	10 MT
½ mile				(*)	(*)
Overpressure, psi	4.1	13	46		
Thermal radiation, cal/cm²	3.8	38	380		
Initial nuclear radiation, rem**	670	6.7×10^3	7.6×10^4		
1 mile				(*)	(*)
Overpressure psi	1.5	4.5	14		
Thermal radiation cal/cm²	0.9	9.1	91		
Initial nuclear radiation, rem**	9.1	91	1100		
2 miles					(*)
Overpressure, psi	<1.0	1.7	5.0	16	
Thermal radiation, cal/cm²	0.2	2.1	21	210	
Initial nuclear radiation, rem**		0.2	1.9	35	
3 miles					
Overpressure, psi		1.0	2.8	8.6	29
Thermal radiation, cal/cm²		0.9	9.0	90	900
Initial nuclear radiation, rem**				<1.0	2.6
5 miles					
Overpressure, psi		<1.0	1.4	4.1	13
Thermal radiation, cal/cm²		<1.0	3.0	30	300
10 miles					
Overpressure, psi			<1.0	1.5	4.5
Thermal radiation, cal/cm²			<1.0	6.6	66
20 miles					
Overpressure, psi				<1.0	1.7
Thermal radiation, cal/cm²				1.4	14
50 miles					
Overpressure, psi					<1.0
Thermal radiation, cal/cm²				<1.0	1.7

*Inside or close to fireball.

**rem—roentgen equivalent man—quantity of any type of ionizing radiation which when absorbed by man produces an effective equivalent to the absorption by man of 1 roentgen of X or gamma radiation.

of a tank as a single unit and almost the entire turret as another.

Cast armor possesses one decided advantage over wrought armor in that it can be poured into almost any desired shape. This permits a tank to be designed with curved surfaces possessing thicknesses and obliquities that provide an efficient utilization of a specified weight of armor to protect a given volume. Furthermore, it eliminates the expense of welding basic armor sections together, and the armor structure is not weakened by the use of welded joints with their relatively inferior shock-resistant properties. Even on vehicles that are essentially made of wrought armor, cast armor is used in gun shields, cupolas, hatch covers, and for small items of complex design.

The disadvantage of cast armor lies principally in the fact that, because of the many variations in the thickness of large armor castings, it is impossible to heat-treat (without the use of an exceedingly difficult preferential heat treatment) to provide optimum ballistic properties throughout. Also, cast armor is usually inherently of more variable quality than wrought armor. Consequently, the resistance to projectile penetration and the resistance to shock displayed by a section of cast armor of a given thickness and obliquity can generally be expected to be slightly inferior to that of wrought homogeneous armor.

Specifications of the metallurgical and mechanical properties of cast homogeneous armor are given in MIL-S-11356D.

2-13.4.1.2 Wrought Homogeneous Armor

Wrought homogeneous armor is also a special alloy steel possessing high hardenability characteristics and a desirable metallurgical structure that is uniform throughout. It is formed into flat plates by rolling or by forging ingots of the steel. The finished plates are then heat-treated to obtain the desired physical and metallurgical properties.

This type of armor is used principally in light tanks, self-propelled artillery, lightly armored vehicles, such as armored infantry vehicles, and sometimes for certain sections of medium and heavy tanks. It is usually employed flat; although it can be formed to a limited extent depending upon its thickness and size. It is convenient to use since it provides flat, smooth surfaces upon which to base the design of the vehicle interior.

Rolled homogeneous armor is somewhat superior to both cast and face-hardened armor in resisting the effects of blast, shock, and penetration. This is partly due to an increase in its ductility, resulting from the hot rolling or forging of the metal, and partly due to the uniformity of its thickness which makes it possible to heat treat the armor to obtain its optimum ballistic properties. Forged armor is usually preferred to rolled armor for sections requiring extremely heavy thicknesses. The physical properties of forged armor are comparable to rolled homogeneous armor since they are both produced by working the metal.

On the disadvantages side, the use of rolled homogeneous armor usually involves the use of many welded joints in the basic construction of an armored vehicle. As a result, it is necessary to contend with the high cost of welding, including the cost of edge preparation, and the relatively lower shock resistance generally associated with welded joints. In addition, while the designer may often find the flatness and uniform thickness of the plates a convenience, he may also discover that these properties sometimes impose serious restrictions on design. In general, each armored surface must now be flat and of the same thickness throughout. This necessitates that compromises be made in vehicle weight and ballistic protection that are not in keeping with the best design.

Specifications of the metallurgical and mechanical properties of wrought homogeneous armor are given in MIL-S-12560B.

2-13.4.1.3 Face-hardened Armor

Face-hardened armor is essentially high-alloy rolled homogeneous steeel armor plate that has been carburized and heat treated to provide it with a very hard face (sometimes over 600 Brinell) and a relatively soft, tough core and back (350 to 450 Brinell). The face-hardened surface is designed to shatter attacking projectiles, thereby reducing their ability to penetrate the armor. However, the extreme hardness, and resultant brittleness, of the plate surface limits its resistance to shock. It is, therefore, employed primarily to gain the advantage of a considerably higher resistance

against penetration by small caliber projectiles than is afforded by an equal weight of homogeneous armor. An overmatching projectile, or one equipped with a cap (see par. 2-13.3.1), can defeat face-hardened armor more readily than it can an equal thickness of homogeneous armor.

This type of armor is considerably more difficult and more expensive to manufacturer than is homogeneous armor. Furthermore, the time required for the carburizing process is appreciable and increases overall production time. This may be a serious factor in critical times. Also, the high carbon content of the hard surface makes welding and repair very difficult. These disadvantages, combined with the low shock resistance, have eliminated the use of face-hardened armor in medium and heavy tanks. It is used primarily in gun shields and in some lightly armored vehicles—although nonferrous armor materials, particularly aluminum alloys, have become increasingly popular for these applications.

2-13.4.2 Aluminum Alloy Armor

Aluminum alloy armor is effective against penetration of small caliber projectiles at very high obliquities. However, projectile impacts cause considerably more spall than occurs on steel armor. Large caliber projectiles impacting on heavy aluminum alloy plates produce cracks in addition to spall. Whenever aluminum alloys appear superior to an equal weight of homogeneous steel armor in resistance to penetration, the superiority can be ascribed to a bulk effect. Since a greater thickness of aluminum alloy is required to give the same protection as an equal weight of steel, a sacrifice in internal space is necessary for tank application (outside dimensions are limited). This condition, together with poor resistance to shock and spall, relegates aluminum alloys to the light armor field at the present time.

Specifications of the metallurgical and mechanical properties of aluminum alloy armor plate are given in MIL-A-46027C, for weldable aluminum armor, and MIL-A-45225A, for forged armor.

2-13.4.3 Other Metals

2-13.4.3.1 Titanium

For practical vehicle design purposes, titanium is a relatively new metal with many outstanding properties. Its corrosion resistance is excellent, and it is extremely strong. It has a strength-to-weight ratio of about 2.6 times that of aluminum and 5.9 times that of steel, based on yield strengths. Nonetheless, it has serious drawbacks. Based strictly on tensile strength ratings, titanium costs about 3½ times as much as aluminum, it is extremely difficult to form, and welding requires special equipment and techniques not presently available to service units. It has also been found that this material tends to spall when subjected to ballistic impacts.

2-13.4.3.2 Magnesium Alloys

Magnesium alloys offer lower resistance to penetration and shock by small arms projectiles than do aluminum alloy plates of the same weight. Like aluminum, magnesium also shows a greater tendency to spall than does homogeneous steel armor, and to date has not been used as tank armor.

2-13.4.4 Plastics and Reinforced Plastics (Ref. par. 2-8.2.4.2)

The general characteristics of reinforced plastics are:

a. Good strength-to-weight ratio
b. Easily formable and fabricated
c. High impact resistance
d. High corrosion resistance
e. Easily repaired
f. Poor ballistic material
g. Expensive to use if very large sections of the vehicle are to be fabricated due to the hand fabrication techniques required
h. Toxic gases are emitted by plastic when hit by projectiles

2-13.4.4.1 Doron II

Doron II is a nonmetallic material that has been developed specifically for ballistic protection. Vests used in Korea used 1/8-in.-thick curved, overlapping pieces of Doron. Tests have indicated that Doron II is very effective in composite armor when used as a back-up material to stop spall and primary fragments.

2-13.4.4.2 Nylon

Bonded nylon has ballistic properties which are quite similar to Doron II, but appears to be

inferior to Doron II in fire resistance. Unbonded nylon requires a thicker layer than Doron II for the same protection.

2-14 UNARMORED VEHICLE HULLS (Ref. par. 1-7.2.)

If one temporarily excludes medium and heavy tanks and tank-type vehicles from this comparison, he will find there is a great deal of similarity between armored and unarmored hulls. Both types provide an enclosure for the cargo and crew and a support for the power plant, drive train, suspension system, and vehicle-mounted weapons and equipment. The amphibious types are faced with the same problems of watertightness, trim, stability, and traction at the land/water interface as are their armored counterparts. Their principal differences stem from the greater weight penalty imposed by the armor of an armored hull.

The unarmored hull provides the designer with somewhat more freedom in his design. Space that would otherwise be taken up by armor is now available to him for vehicle components, weapons, or cargo space. The vehicle can be made smaller and lighter as a result. Reduced hull weight affects the size of the power plant and suspension system, all of which contribute to achieving a smaller, lighter vehicle. Access doors for personnel and for maintenance functions, no longer required to resist ballistic impacts and blast, can be made larger to facilitate loading and maintenance operations. Similarly, the releasing of ballistic protection requirements permits improved driver visibility and eases the hull and engine compartment cooling problems.

An armored vehicle hull generally possesses considerable reserve strength and stiffness to road loads in the strength of the armor needed for ballistic protection. This permits the designer a degree of inaccuracy in predicting the magnitude of the road loads anticipated for the vehicle. The unarmored vehicle, however, lacks this strength reserve and, therefore, requires a more accurate evaluation of the anticipated loads and a thorough analysis of the forces experienced by all parts of the hull and their effects upon the structure, both immediate and long-term effects.

In the design of an armored hull, major considerations are the degree of protection required, the cubic space required to be enclosed, and the maximum allowable weight. In the design of an unarmored hull, the major considerations are mimimum utilization of space, and a high level of durability. Toward these ends the unarmored hull designer makes use of high strength steel or aluminum alloy plates, monocoque or semimonocoque construction, welded joints, and even fabrics and plastics where a weight saving can be accomplished.

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CHAPTER 3
COMPONENTS ASSOCIATED WITH BODY AND HULL DESIGN*

SECTION I—INSULATION

3-1 THERMAL INSULATION

3-1.1 PRINCIPLES OF HEAT DISSEMINATION

Heat is transferred from warm to cold bodies by processes of convection, conduction, radiation, or by a combination of these three. All three heat transfer processes must be considered in the design of military vehicles, not only from the standpoint of insulation requirements but also in calculating vehicle heating, cooling, and ventilating loads.

Convection is the transfer of heat by moving matter; for example, a liquid or a gas flowing over two bodies of different temperature absorbs heat from the hotter and loses it to the colder. Heat transfer by convection is independent of the nature of the surfaces over which the conveying agent moves. The amount of heat transferred by convection per unit of time is affected by the velocity of the moving medium, the area and shape of the surfaces exposed to the moving medium, and the temperature difference between the moving medium and the surface over which it flows.

Conduction is the transfer of heat through matter, from one particle of the material to an adjoining particle, with no apparent motion of the matter involved.

An example of conduction is the passage of heat through a metal bar when one end is held in a flame. The rate of flow varies with the character of the material, the distance through which the flow takes place, and the temperature difference that is maintained. Some materials permit a rapid transmission of heat by conduction and soon become uniform in temperature, while other materials resist the flow of heat through them and, therefore, act as insulators.

Radiation is the transfer of heat through space without the apparent involvement of intervening matter. Heat transferred by radiation passes through space in straight lines following the laws of travel of light. The classic example of this type of heat transfer is the transmission of heat from the sun to the earth. Some materials are absorbers and others are reflectors of radiant heat. Materials generally have the ability to emit the same amount of radiant heat that they can absorb under reversed conditions of temperature difference. A few materials have the property of allowing radiant heat to pass through them without absorbing any of the heat transmitted. Other materials diffuse radiant heat in all directions. The transmission of radiant heat varies inversely as the square of the distance between the source of radiation and the absorbing surface. Heat emission by radiation is dependent upon the nature of the radiating surface, its absolute temperature, and the temperature difference maintained between the radiating and receiving bodies. The heat transmission varies as the difference of the fourth powers of the absolute temperatures of the radiating and receiving bodies (Stefan-Boltzmann Law).

Heat flow can be considered to be either time dependent or independent of time (time constant). The former is referred to as unsteady state flow and the latter as steady state flow. In most military vehicle applications, steady state flow conditions are assumed. Conditions of unsteady state flow exist during the thermal radiation phase associated with the initial effects of a nuclear detonation.

3-1.1.1 Convection

Therefore, assuming steady state flow conditions, heat transferred through convection from a surface can be expressed as

$$Q_h = hA \cdot \Delta t \tag{3-1}$$

*Written by Ernest Bergmann and Rudolph J. Zastera of the IIT Research Institute, Chicago, Illinois.

where

Q_h = steady state heat flow due to convection, Btu/hr

h = film coefficient of heat transfer, Btu/hr-ft^2-°F

A = surface area in contact with fluid, ft^2

Δt = temperature difference between fluid and surface, °F

3-1.1.2 Conduction

Similarly, heat transfered through a thickness of material by means of conduction can be expressed as

$$Q_k = kA \frac{\Delta_x t}{x} \tag{3-2}$$

where

Q_k = steady state heat flow due to conduction, Btu/hr

k = thermal conductivity of the material, Btu/hr-ft^2-°F/ft or Btu/hr-ft-°F (see following paragraph)

A = area perpendicular to heat flow, ft^2

$\Delta_x t$ = temperature difference across thickness of material, °F

x = thickness of material parallel to heat flow, ft

The coefficient of thermal conductivity k in Eq. 3-2 is the amount of heat, in British thermal units per hour, that is transmitted by conduction from one square foot of area per unit temperature gradient; the units of the last term being of per foot of material thickness. Thus, the units of k are Btu/hr-ft^2-°F/ft, which can be reduced to Btu/hr-ft-°F. Tabulated values of k for various materials are often based upon a unit temperature gradient of °F per inch of thickness rather than °F per foot of thickness. In such cases, convert the value of x in Eq. 3-2 to inches, or else decrease the value of k by a factor of 12.

3-1.1.3 Radiation

A general expression for the transmission of heat by means of radiation per unit of surface according to the Stefan-Boltzmann Law is

$$Q_r = K\left(T_1^4 - T_2^4\right) \tag{3-3}$$

where

Q_r = steady state heat radiated, Btu/hr-ft^2

K = a coefficient that includes a radiation constant, a relative-position factor for the emitting and receiving surfaces, and absorption and emission characteristics of the two surfaces

T_1 = temperature of heat radiating body, °F abs

T_2 = temperature of heat receiving body, °F abs

The constants h, k, and K which appear in Eqs. 3-1, 3-2, and 3-3 are difficult to determine accurately for a number of reasons. The film coefficient of heat transfer h is a function of thermal conductivity, viscosity, density, specific heat, velocity, shape and arrangement of heating surface, direction of flow relative to heating surface, turbulence, and the temperature level. Thermal conductivity k is dependent upon the material, its density (lb/ft^3), and the temperature level. The radiation coefficient K depends upon such additional factors as surface roughness, color, and texture of the emitting and receiving surfaces, the materials involved, and the temperature level. Furthermore, in an actual environment, the pure conditions seldom exist; and considerable interaction takes place among the three modes of heat transfer. Where one mode predominates, however, one of the three equations given can be applied. Tabulated values for the constants are given in many references on heat transfer. Refs. 1 through 7 are particularily useful.

3-1.1.4 Overall Coefficient of Heat Transmission

The calculation of heat transmitted by convection, conduction, or radiation involves the use of surface temperatures, and these are often difficult to obtain. A method used to calculate heat flow through walls uses an overall coefficient of heat transmission U that includes the relative effects of the three modes of heat transfer, the relative placement of materials, and is used in conjunction with the difference in the

air temperatures that exist directly adjacent to the wall section under consideration. The equation for heat transmission through a wall, based upon this coefficient, is

$$H = UA\,(t - t_o) \tag{3-4}$$

where

H	=	steady state heat flow, Btu/hr
U	=	overall coefficient of heat transmission, Btu/hr-ft^2-°F
A	=	area of wall (floor, roof, etc.), ft^2
t	=	inside air temperature, °F
t_o	=	outside air temperature, °F

The overall coefficient of heat transmission U may be obtained experimentally or by calculations based upon known data for the materials involved. Since the experimental determination is both laborious and expensive, most tabulated values of U for various materials, used alone or in combinations with other materials, are determined by calculation; and experimental methods are used to verify selected values and to supply basic data for the calculations. Values of U for composite walls of buildings are given in Refs. 1-7. The typical equation for determining U is of the form

$$U = \frac{1}{\dfrac{1}{f_i} + \dfrac{1}{f_o} + \dfrac{x}{k}} \tag{3-5}$$

where

f_i	=	combined effects coefficient of convection and radiation at the warmer surface of the wall, Btu/hr-ft^2-°F
f_o	=	combined effects coefficient of convection and radiation at the wall surface of lower temperature, Btu/hr-ft^2-°F
x	=	material thickness, ft
k	=	thermal conductivity of the material, Btu/hr-ft-°F

The same cautionary note regarding the units of k and x made in the discussion following Eq. 3-2 also apply here.

3-1.1.5 Transmission Coefficients for Compound Walls

In situations where the wall under consideration is made up of layers of different materials, Eq. 3-5 must be extended to include the effects of the additional materials and will appear as

$$U = \frac{1}{\dfrac{1}{f_i} + \dfrac{1}{f_o} + \dfrac{x_1}{k_1} + \dfrac{x_2}{k_2} + \dfrac{x_3}{k_3} + \cdots + \dfrac{x_n}{k_n}} \tag{3-6}$$

where

x_1, x_2, x_3, and x_n	=	respective thickness layers 1, 2, 3, and n, ft
k_1, k_2, k_3, and k_n,	=	respective thermal conductivities of materials in layers 1, 2, 3, and n

When layers of air are included between layers of a wall as part of the wall construction, the conductance of the air space must be added to the equation. Conductance C is defined as the amount of heat, in Btu per hour, that passes through one square foot of area of a material of the thickness and arrangement stated, per one degree Fahrenheit temperature difference of the surface temperatures. It is always entered into the equation for U as $1/C$. In the equation which follows, the term a designates the conductance of a particular air space. Thus, assuming there is an air space between layers 1 and 2, Eq. 3-6 would be modified to

$$U = \frac{1}{\dfrac{1}{f_i} + \dfrac{1}{f_o} + \dfrac{x_1}{k_1} + \dfrac{1}{a} + \dfrac{x_2}{k_2} + \dfrac{x_3}{k_3} + \cdots + \dfrac{x_n}{k_n}} \tag{3-7}$$

The conductances a of air spaces depend upon (1) the mean temperature of the confined air, (2) the amount of air movement occurring, (3) the width of the air space, (4) the nature of the bounding surfaces as regards their emissivity and reflectivity, and (5) whether the air space is positioned vertically or horizontally. In general, there is little increase in conductances for air

spaces wider than one inch for a given mean air temperature; and the conductances of air space of fixed width increases as its mean air temperature increases. For vertical air spaces one inch wide and greater and bounded by nonreflective surfaces, the conductance varies approximately linearly from about 1.02, at a mean temperature of 20°F, to about 1.7, at a mean temperature of 150°F.

Values of the conductances f_i vary from about 1.4 to 2.1 for still air conditions—depending upon the materials, surface conditions, and mean temperatures. A commonly accepted value of f_i for all materials in still air is 1.65. Conductances f_o vary greatly with air velocity, surface roughness, and mean temperature. An average value found to be generally acceptable for all materials, with the average wind velocity over their surfaces of 15 mph, is about 6.0. Other values can be calculated from the following equations in which v is the velocity of the air in mph.

$f_o = 1.4 + 0.28\,v$; for very smooth surfaces such as polished glass and bright finished cold rolled sheets (3-8)

$f_o = 1.6 + 0.3\,v$; for smooth surfaces such as cold rolled sheet metal panels, smooth plastic panels, and smooth white painted surfaces (3-9)

$f_o = 2.0 + 0.4\,v$; for average surfaces such as smooth castings and forgings, hot rolled plates, and smooth concrete (3-10)

$f_o = 2.1 + 0.5\,v$; for rough surfaces such as rough castings, heavily scaled and rusted plates, rough textured surfaces, rough concrete, and stucco (3-11)

Thermal coefficients k for various materials are given in Tables 3-1 and 3-2. Other data of this type along with more detailed explanation of this general subject are given in Refs. 1-7, and in Refs. 10 and 11. Refs. 10 and 11 are especially comprehensive and comprise 5 and 6 volumes, respectively. Practically every known material is cataloged in these volumes.

3-1.2 PRINCIPLES OF THERMAL INSULATION

Thermal insulation is principally concerned with retarding the flow of heat due to conduction and radiation by providing barriers in the path of the heat flow. Barriers, or insulators, to heat flow due to conduction are materials that have very low thermal conductivities. They are generally characterized by a large number of air cells or air spaces enmeshed within them which gives these materials a low bulk density (lb-ft^3). The presence of moisture in the insulation, however, tends to increase the coefficient of conductivity and lower the insulating value. Barriers to heat flow due to radiation are materials that have smooth, bright, highly reflective surfaces. An example of these are thin sheets or foils of certain metals such as tin foil or aluminum foil. These materials lose some of their capacity to repel radiant heat when their surfaces become tarnished or dirty.

The most efficient insulating medium is an evacuated space separating the primary area from its surrounding environment. Since air molecules remaining in the evacuated space are the only medium that can conduct heat, the efficiency of this insulating jacket depends upon the degree of vacuum that is maintained. However, the relationship between the thermal conductivity of a partially evacuated air space and its absolute pressure is not linear. The partial evacuation of an air space has only a very negligible effect upon its thermal conductivity above a pressure of about 100 torr (one torr= pressure of 1 mm Hg abs at 0 °C). Below 100 torr, the conductivity decreases quite rapidly with decreasing pressure down to approximately 0.01 torr. Below this point, the conductivity curve begins to level off. An appreciable reduction in thermal conductivity (approximately one-tenth the value at atmospheric pressure) is attained at about 0.15 torr. The lowest values are reached below 0.001 torr; although pressure reductions below 0.001 torr have almost no effect on further reducing heat flux. Thermal conductivity at this point (0.001 torr) is approximately 0.0003 Btu/hr-ft-°F (Ref. 12). This is shown graphically in Fig. 3-1. When the bounding surfaces of the evacuated space are smooth, bright, and highly reflective, heat transmission across the space due to radiation is also

TABLE 3-1 THERMAL CONDUCTIVITIES OF VARIOUS METALS AT SPECIFIED TEMPERATURES*

Metal	Temp, °F	k, Btu/hr-ft-°F
Aluminum	64	117
	212	119
Aluminum Alloys		
2014, 2025, 2618	80	90
2024	200	80
	290	90
	400	100
	600	105
	800	100
2017, 5454	80	78
5052, 359, 195, 40-E	77	80
6061 (H. T.)**	77	90
6061 (H. T. & aged)	77	96
6062 (H.T.)**	77	98
6062 (H. T. & aged)	77	104
7075	100	78
	200	80
	300	86
	400	100
	460 - 700	120
	800	99
7079, 2219, 354	77	74
7178, 5086	80	72
6151	77	100
C355, 355, A356	77	88
356, 357	77	88
220, 2020	77	51
5083, 5456	80	68
Cadmium	64	53.7
Copper	64	224
Copper Alloys		
Coml brz† 90% Cu	-	109
Red brs‡, 85% Cu	-	92
Low brs‡, 80% Cu	-	81
Cartridge brs‡, 70% Cu	-	71
Yellow brs‡, 65% Cu	-	67
Leaded brs‡,	-	67
Admiralty brs‡,	-	64
Naval brs‡	-	67
Mn brz†	-	61
Al brs‡	-	58
Ph brz†, 5% Sn (A)	-	47
Ph brz†, 8% Sn (C)	-	36
Ph brz†, 10% Sn (D)	-	29
Ph brz†, 1.25% Sn (E)	-	120
Al brz†, 5% Al	-	48
Al brz†, 8% Al	-	42
Cupronickel, 30% Ni	-	17
Cupronickel, 10% Ni	-	26
Nickel-Silver, 18% Ni	-	17 - 19
High-Silicon brz†	-	21
Low-Silicon brz†	-	33
Beryllium Cu	-	46
Beryllium Cu (H.T.)**	-	47 - 64

Metal	Temp, °F	k, Btu/hr-ft-°F
Iron, pure	64	39
	212	36.6
Iron, wrought	64	34.9
	212	34.6
Iron, cast	129	27.6
	216	26.8
Lead	64	20.1
	212	19.8
Magnesium	32 - 212	92.0
Magnesium Alloys		
AZ31B	212 - 572	56
AZ61A	212 - 572	46
AZ63A, AZ80A, AZ81A	212 - 572	44
AZ91C (cast)	212 - 572	41
AZ92A (cast)	212 - 572	39
EZ33A (cast)	80	58
HK31A	68	60
HZ32A (cast)	80	66.8
M1A	212 - 572	73
ZH62A	80	62.9
ZK51A	80	65
ZK68 (cond F)	68	68
ZK60A (cond T5)	68	70
HM21A	212 - 572	80
HM31A	212 - 572	61
QE22A (cast)	70	59
LA141A	75	25
Nickel	64	36
	212	34
Steel (AISI 1025)	32	30.0
Steel (1% carbon)	64	26.2
Steel Alloy (AISI 4130, 4140, 4340, 8630, 8735, 8740)	32	22.0
Steel (13 Cr, 0.2 Ni)	932	16.6
Steel (18 Cr, 8 Ni)	100	8.5
	1600	15.9
Steel (23 Cr, 12 Ni)	932	10.8
Steel, Heat Resistant Fe-Cr-Ni Base Alloys		
A-286	100	7.15
	1200	14.3
N-155	70	7.08
	800	10.9
	1200	12.8
W-545	300	6.6
	1200	10.7
	1600	13.0
Ni-Base Alloys		
Inconel 600	80 - 212	8.6
	1472	13.1
Hastelloy B	392	7.1
	752	8.0
	1112	9.5

Metal	Temp, °F	k, Btu/hr-ft-°F
Ni-Base Alloys (cont'd.)		
Hastelloy X	68	5.6
	1652	15.8
M-252	70	6.8
	1000	10.5
	1500	12.4
Rene 41	200	6.8
	1600	13.3
Inconel X-750	200	9.0
	1200	16.9
	1600	20.0
Udimet 500	100	6.5
	200	6.9
	400	7.7
	800	9.6
	1600	14.1
Co-Base Alloys		
L-605	200	6.2
	1500	14.3
	1700	15.9
Tin	64	36
	212	34
Titanium, pure	65	10.2
	100	10.6
	200	11.3
	280 - 800	11.5
	1000	11.6
	1200	11.8
	1400	12
Titanium Alloys		
Ti-5Al-2.55 Sn	100	4.6
	200	4.8
	400	5.3
	600	6.1
	1400	10.0
Ti-8Al-1Mo-1V	80	3.5
Ti-8Mn	80	6.3
	1400	11.3
Ti-4A1-4Mn	80	4.0
	100	4.1
	200	4.4
	1600	11.1
Ti-4 Al-3Mo-1V	80	3.9
Ti-5 Al-1.5Fe-1.4Cr-1.2Mo	100	4.8
	400	5.6
	800	7
Ti-6 Al-4V	100	4.2
	400	5.0
	1000	8.0
Ti-6Al-6V-2Sn	80	4.2
Ti-13V-11Cr-3Al	80	4.0
Zinc	64	65
	212	64

*Data were collected from various sources, but mainly from Refs. 5, 7, and 8. The variation of k is substantially linear between the temperatures given.

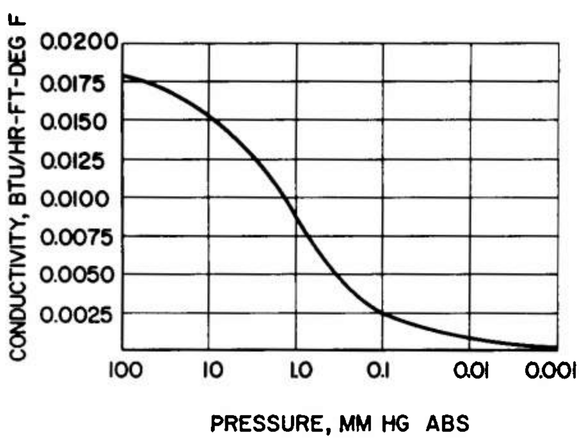
**H.T. = heat treated †brz = bronze ‡brs = brass

TABLE 3-2 THERMAL CONDUCTIVITIES OF VARIOUS SOLIDS*

Material	*Bulk Density*, lb/ft^3	*Temp*, °F	*k*, $\frac{Btu}{hr\text{-}ft\text{-}°F}$	*Material*	*Bulk Density*, lb/ft^3	*Temp*, °F	*k*, $\frac{Btu}{hr\text{-}ft\text{-}°F}$
Asbestos board	123	86	0.225	Mineral wool board (Rockcork)	14.3	100	0.024
Asbestos millboard	60.5	86	0.070	Pearlite, Arizona	9.1	112	0.035
Asbestos paper, laminated	22	100	0.038	Polystyrene, expanded (Styrofoam)	1.7	—	0.021
Asbestos paper, corrugated	16	100	0.031	Pumice, powdered	49	300	0.11
Asbestos wool	25	212	0.058	Quartz, perpendicular to C-axis	—	-300	12.5
Balsa wood	20.0	—	0.048			0	4.3
Balsa wood	7.3	—	0.028			300	2.3
Balsam wool	3.6	70	0.021	Quartz, parallel to C-axis	—	-300	25.0
Celotex	13.2	—	0.028			0	8.3
Concrete, sand & gravel	142	75	1.05			300	4.2
Concrete, cinder	97	75	0.41	Redwood bark, shredded, loose	4.0	100	0.025
Corkboard, pure	14.0 to	—	0.028 to	Rockwool, loose	7.0	117	0.024
	10.6		0.025	Rubber, hard	74.3	100	0.092
Corkboard	6.9	100	0.022	Rubber, soft, vulcanized	68.6	86	0.08
Cork, granulated	5.4	23	0.028	Rubber, expanded (Rubatex)	4.9	100	0.018
Cotton wool	5.0	100	0.035	Sand, dry	94.8	68	0.188
Fiberglas block, PF612	2.5	100	0.023	Sawdust, dry	13.4	68	0.042
Fiberglas block, PF614	4.25	100	0.021	Shavings, wood	—	—	0.059
Fiberglas block, PF617	9	100	0.020	Silica, fused	—	200	0.83
Fiberglas with asphalt				Silica gel, powder	32.5	131	0.049
coating, board	11	100	0.023	Silica aerogel powder (Santocel)	5.3	100	0.013
Foamglas	10.6	100	0.036	Sil-O-Cel powder	10.6	86	0.026
Glass, pyrex	139	200	0.59	Thermofelt (jute & asbestos)	10.0	—	0.031
Glass, soda lime	—	100	0.59	Thermofelt (hair & asbestos)	7.8	—	0.023
Glass wool	3.25	100	0.022	Thermofill (flaked gypsum)	34.0	—	0.019
Hairfelt	11.0	86	0.022	Urethane foam, fluorocarbon blown	1.5 to 2	—	0.0093
Hairinsul (hair & jute)	6.1	—	0.022	Urethane foam, carbon dioxide blown	1.5 to 2	—	0.021
Insulite	16.9	100	0.028	Vermiculite, loose	8.2	60	0.038
Kaolin wool	10.6	800	0.059	Wood, dry, across grain**			
Kapok	1.6	100	0.019	Douglas fir	29	85	0.063
Kaylo (hydrous calcium silicate)	11	100	0.032	Maple, sugar	43	85	0.094
Leather, sole	62.4	—	0.092	Oak, red	42	85	0.099
Magnesia, 85%	12	100	0.029	Pine, white	25	85	0.060
Magnesia, rigid (15% asbestos)	19.3	—	0.043	Spruce	21	85	0.052
Masonite	19.8	—	0.027	Zirconia grain	113	600	0.11
Micro-quartz fiber	3	100	0.021				
Mica	122	—	0.25				

*Data compiled from various sources but mainly from Refs. 5, 7, and 9.

**With heat flow parallel to the grain, k may be 2 to 3 times greater than with heat flow perpendicular to grain.



Data based upon experiments with nitrogen. (Courtesy of *Cryogenic Engineering Conference Proceedings*, 1962, Ref. 12)

Figure 3-1. Apparent Mean Thermal Conductivity vs Pressure Between 76° and 300°K

minimized. These are the principles of the common vacuum bottle.

The vacuum bottle concept is not applied to military vehicles at the present time partly because of the design problems involved in maintaining high vacuums, particularly in a military environment, and partly because of the nature of general military cargo which doesn't require superinsulating techniques. A possible exception to this may be the tank bodies of tank trucks and tank trailers designed for transporting cryogenic materials such as liquefied hydrogen, oxygen, nitrogen, etc. The civilian counterparts of these vehicles do employ superinsulating techniques. A case in point is six "jumbo" railway tank cars for the transcontinental shipment of liquid hydrogen which have an evaporation loss rate of less than 0.3 percent of capacity per day. The car is 75 ft long and has a capacity of 28,300 gal or 16,500 lb of liquid hydrogen. Special and unique design features were necessary because of the extremely low boiling point of liquid hydrogen, its low heat of vaporization, and its extreme flammability. A detailed discussion of this design is given in Ref. 13.

Since a vacuum must be maintained at a pressure of 100 torr or less to be significantly more effective as a thermal insulator than is dry, still air, it is generally more practical and more economical to dispense with the vacuum and use air spaces for insulation. In order to be effective, however, the air spaces should be at least 0.7 in. thick (one inch is preferable). Increasing the thickness of the air space above one inch results in a negligible improvement in its insulating properties.

The use of dead air spaces for thermal insulation is generally wasteful of space, particularly in vehicle design where space is generally at a premium. A more efficient design makes use of structural materials that have low thermal conductivity, uses highly reflective surfaces where practical to minimize heat transmission by means of radiation, and places additional insulating materials in the path of the heat flow by lining the enclosure with low conductivity material. Since moisture has a deterimental effect on the thermal conductivity of many materials, these should be avoided or else used with suitable vapor barriers.

Vapor barriers are materials that resist the passage of water vapor (metal foils, certain plastic foils, etc.). These should be installed on the warm side of the insulation if the temperature and humidity conditions are such that condensation might take place within the insulation.

3-1.3 APPLICATIONS

Because a degree of austerity is a required characteristic of military vehicles, they are not generally provided with thermal insulation. Thermal insulation is provided when it will improve a vehicle's mission capabilities without degrading its functional characteristics or its reliability. It is common on van-type vehicles (including van-type trailers), tank-type vehicles designed for the transport of water and cryogenic fluids, and on special vehicles designed for extreme cold weather operations.

Combat vehicles are generally not insulated since every pound of excess weight and every cubic inch of increased volume detract from the vehicle's maneuverability and increase its vulnerability to enemy fire. Crew comfort in combat vehicles is generally accomplished by heating and ventilating the crew and passenger compartments; although, thermal insulation is sometimes applied between the engine and crew compartments. Occasionally, a degree of thermal insulation is obtained indirectly, as from the use of plastic or spaced armor, or from nuclear radiation shields.

The amount and type of insulation required depends upon each application. Tank bodies for transporting water require some insulation to resist freezing temperature in cold climates and to prolong the water's potability in warm climates. Plastic tanks have been used to good advantage for this purpose. The plastics used have a sufficiently low thermal conductivity in comparison to metals that no additional insulation is required. The construction of van-type bodies, on the other hand, utilizes materials and techniques which generally require the addition of insulating materials when reasonable interior temperatures must be maintained:

3-1.4 MATERIALS

A large variety of materials that have low thermal conductivities, both natural and manmade, are available to the designer. Most natural materials, however, particularly the organic types, absorb moisture, promote fungus growth, and harbor insects and vermin which make them unsuitable for use in a military environment unless they are processed to overcome these undesirable characteristics. Processed insulating materials are available as loose fill, blankets, batts, boards, sheets, and various foams.

Organic insulating materials—such as fiberboards, corkboard, balsa wood, or paper—should not be used above 212°F. For temperatures up to 500°F, 85 percent magnesia rock, rock and glass wool, and asbestos paper with air cells are frequently used. For higher temperatures, refractory type materials are required such as diatomaceous silica, slag and rock wools, and bricks of porous refractory materials. The paragraphs which follow discuss some of the more commonly used insulating materials.

a. *Mineral Wools.* These are fibrous substances that are made from a variety of materials such as limestone, shale, slag, or silica sand. The raw material is heated to its molten state and then blown by air or steam into a fibrous form and allowed to become deposited as a loose wool. This wool is then fabricated into batts, blankets, or boards.

b. *Fiber Boards.* These comprise a varied assortment of rigid, semirigid, and flexible materials manufactured by combining various fibers with suitable binders and compressing them into slabs or boards of various thicknesses. During processing, the material is treated to repel vermin and to make it resistant to moisture and fire. Typical materials used for this type of insulation are wood fibers, tree bark, shavings, wood pulp, sawdust, marine plants (eel grass), sugar cane, kapok (fibers from seed pods of the kapok tree), animal hair, and others.

c. *Cork.* This is the outer bark of the cork oak. It is of light weight (10 to 12 lb/ft^3), is highly compressible, possesses a high degree of permanent elasticity, is a very poor conductor of heat, is waterproof, unaffected by moisture, and is slow burning. For insulation applications, pure corkboard is available in sheets 12 X 36 in. and in a choice of thicknesses up to 6 in. The natural gums and resins within the cork are adequate to hold the material together after it is properly baked, so that no artificial binders are needed. Granulated cork is also used for insulation as a filling material between confining walls.

d. *Asbestos.* This is a heat-resisting fibrous mineral that has a large variety of applications because of its incombustibility, low heat and electrical conductivity, and its resistance to the action of most chemical agents. It is spun and woven into yarn, rope, and cloth; is formed into asbestos felt, paper, roll board, and mill board for insulation; is mixed with portland cement to make asbestos wood and rigid panels; and it constitutes either the principal ingredient or the essential reinforcing material for most insulations intended for application in temperatures ranging from 150° to 1900°F. Among the latter are such materials as laminated-felt types consisting of successive layers of thin felts, the felted-fiber types (asbestos fiber and bonding materials), and the air-cell types made up of successive layers of plain and corrugated asbestos paper.

e. *Expanded Minerals.* These comprise materials, such as perlite and vermiculite, that are made by expanding certain moisture-containing minerals. When these minerals are crushed and then heated rapidly to a sufficient temperature, they expand to many times their original volume (a process much like the popping of corn) to form a lightweight, highly effective thermal insulating material. Perlite is made from a moisture-containing volcanic rock found in large deposits throughout

the world; vermiculite is made from mica.

f. *Plastics.* Plastic insulating materials comprise the cellular plastics commonly referred to as "foamed" or "expanded" plastics. These fall into two general types with regard to their structure, namely, the closed-cell, and the open-cell. In the closed-cell type, each spherically shaped cell is completely enclosed by a wall of plastic making the material impervious to liquids and, therefore, also suitable for flotation devices. In the open-cell type, the individual cells are interconnected.

Cellular plastics, or foams, are made in a wide range of densities from 0.1 to 75 lb/ft^3 and in rigid, semi-rigid, and flexible form. Nine general types of plastics are used for making foams. These are cellulose acetate, epoxies, phenolformaldehyde, polyethylene, polystyrene, silicones, urea-formaldehyde, urethanes, and vinyls. They are produced in the form of slabs, blocks, boards, sheets, molded and extruded shapes, and as sprayed coatings. In addition, several types can be foamed in place or packed in place into an existing cavity. Table 3-3 lists the densities, thermal conductivities, and service temperatures of various foamed plastics. For more detailed information consult Refs. 14 and 15.

3-2 ACOUSTIC INSULATION

3-2.1 PRINCIPLES OF SOUND AND NOISE

Sound results from a series of alternating pressure variations in the atmosphere (or in any suitable medium) and becomes audible to the average person when the rate, or frequency, of these variations is within the range of about 16 to 16,000 cps. A continuous series of such variations occurring at only a single frequency results in a pure tone. Other frequencies can be superimposed upon the already vibrating air. When these superimposed frequencies are simple multiples of the first frequency, they produce a series of harmonics of the fundamental tone, and the result is a musical sound. However, if the range of the superimposed frequencies extends throughout and beyond the audible range, the resulting sound is unpleasant to the ear and is referred to as noise. Typical examples of noise are the clashing of gear teeth, the vibrations induced by unbalance in rotating machinery, the continuing explosions within an internal combustion engine, the roar of escaping steam, the hiss of compressed air, and the whine of an internal combustion turbine.

In addition to the frequencies that cause the fundamental excitations, noise also contain the resonant frequencies of the various vibrating surfaces that are involved, such as the gear teeth, the machine frame, piping, turbine blades, etc. In general, small surfaces are sources of high-frequency noises, and large surfaces radiate low frequencies. Irregular excitations of vibrating surfaces—as by road shocks, ballistic impacts, or scraping—result in noises whose frequency spectrums are chiefly determined by the vibrational properties of the surfaces involved.

In a broad sense, noise embraces all sounds that are audibly unpleasant, and this includes those that are unpleasant by virtue of their loudness or pitch (frequency) as well as those whose quality is unpleasant. Thus, the clamor of a high intensity bell (95 dB) is unpleasant even though its basic sound is a pleasant musical tone. Similarly, an extremely high pitched whistle may become annoying when sounded continuously even though it may be of a pure tone of average intensity.

All sounds have three characteristics; namely, quality, intensity, and loudness. *Quality* is a subjective attribute of sound by which equally loud sounds can be distinguished as different kinds. Its basis is the distribution of energy among the various frequencies that constitute the sound. Differences in quality affect both the sensation of loudness of noise and its psychological annoyance. Shrill, high-pitched, and irregular sounds are generally judged less pleasant than low-pitched and regular sounds.

Intensity refers to the sound power transmitted through a unit area of the pressure wave front and is usually expressed in decibels by comparing it to a standard reference intensity. The decibel is defined as 20 times the logarithm to the base 10 of the ratio of the actual sound pressure to the reference sound pressure (0.0002 microbar is most commonly used for airborne sound). Table 3-4 gives the sound pressure levels of some common sounds and environments.

Loudness is another subjective attribute of sound (and noise) which combines the effects of sound quality and the differences in acuity of

TABLE 3-3 DENSITY, THERMAL CONDUCTIVITY, AND MAXIMUM SERVICE TEMPERATURE OF FOAMED PLASTICS‡

Type	*Density,* lb/ft³		*k,** Btu/hr-ft-°F		*Maximum Service Temperature,* °F	
	High†	*Low*†	*High*†	*Low*†	*High*†	*Low*†
Butadiene-Acrylonitrile foams	25	10	0.025	0.021	210	—
Butadiene-Styrene foams	4.37	—	0.018	—	160	—
Cellulose Acetate, rigid, prefoamed	6.24	3.7	0.027	0.025	350	200
Epoxy, rigid, prefoamed	20	1.8	0.02	0.009	200	160
Natural rubber foams	7	6	0.025	0.021	160	—
Neoprene foams	30	10	0.029	0.021	180	—
Phenolic, rigid, foamed in place	22	0.3	0.023	0.017	300	200
Polyethylene foams (low density)	2.6	1.9	0.33	0.029	160	130
Polystyrene, rigid, foamed in place	20	10	0.02	—	185	—
Polystyrene, rigid, prefoamed	6	1	0.023	0.019	175	155
Silicone foam, rigid	16	3.5	0.03	0.025	650	500
Urea-Formaldehyde foams	3.7	0.5	0.02	0.017	350	150
Urethane foam, flexible	20	1	0.025**	0.019**	275	200
Urethane, rigid, foamed in place	25	0.5	0.03	0.01	400	200
Vinyl foams, flexible	75	4	—	—	225	125

*k = thermal conductivity in Btu/hr-ft²-°F/ft.
†"High-Low" values represent limits of a range of typical values.
**Values for density of 2 lb/ft³.
‡Compiled principally from Refs. 14 and 15.

TABLE 3-4 TYPICAL SOUND PRESSURE LEVELS OF SOME COMMON SOUNDS AND ACOUSTICAL ENVIRONMENTS

Sound Pressure		*Description*	
dB	*dyne / cm²*		
0 10	0.0002 0.001	Very faint	Threshold of audibility Soundproof room
20 30	0.002 0.01	Faint	Whisper Rustle of leaves Quiet conversation Quiet auditorium
40 50	0.02 0.1	Moderate	Quiet home Quiet radio Average conversation Average office
60 70	0.2 1.0	Loud	Noisy home Average factory Average radio Average street noise, noisy typewriter
80 90	2 10	Very loud	Noisy office Police whistle, loud speech Heavy street traffic Noisy factory
100 110	20 100	Deafening	Very loud street noise Boiler factory Elevated trains Nearby riveter
120 130	200 1,000	Very deafening	Loud thunder and artillery fire Jet engines Threshold of feeling Ear discomfort
140 150 160	2,000 10,000 20,000	In-tolerable	Jet engines

the human ear to various frequencies. The human ear is most sensitive to frequencies in the 500 to 5,000 cps range with maxium acuity at about 3,500 cps. Thus, sounds of equal intensity but differing in frequency are judged to differ in loudness. Similarly, unpleasant noise, being more audibly conspicuous because of the frequency band they encompass, are also judged to be louder than their intensities indicate. The loudness level of a sound is evaluated quantitatively in phons. This unit is defined as the pressure level (in decibels) of a pure tone of 1,000 cps that is judged to sound as loud as the sound being investigated. Fig. 3-2 shows lines of equal loudness for the audible frequency and intensity range. The lowest line represents the lower threshold of audibility, and the topmost line represents the threshold at which sound perception merges into a feeling sensation within the ear.

3-2.2 EFFECTS OF NOISE

Excessive noise can produce adverse effects on vehicle structures, on equipment, and on personnel. It induces and augments vibrations in

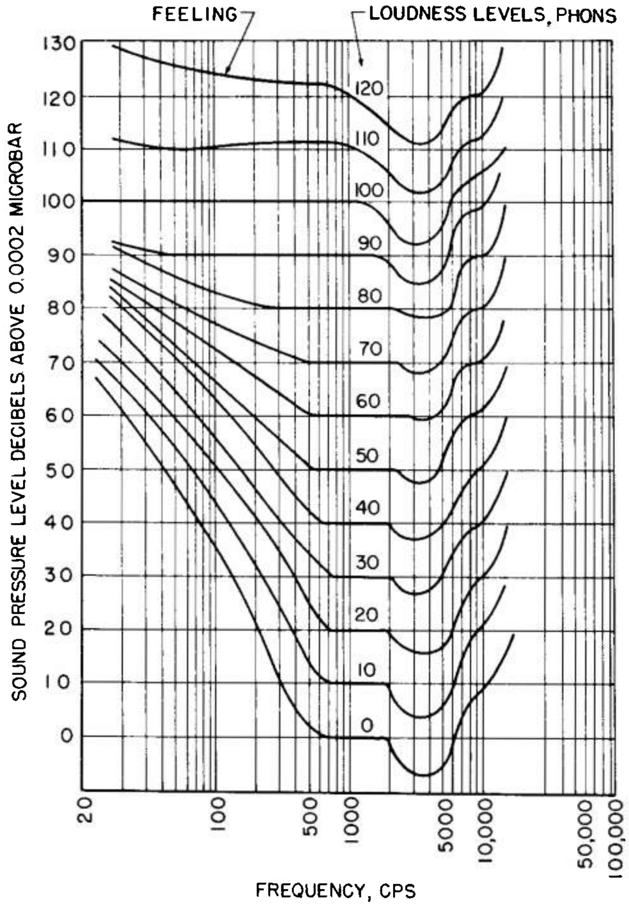


Figure 3-2. Loudness Levels of Various Frequencies and Intensities

structural panels, thus enhancing the undesirable effects associated with vibrations. Inadequate attention to proper noise control during the initial design of a vehicle may result in a noise environment so severe that excess weight penalties will be incurred to remedy the condition. This may be particularly true in the case of lightweight, thin-skinned, hull-type vehicles. Furthermore, certain acoustical noises may cause intermittent operation of electronic equipment and, at some levels, may cause actual permanent damage. Malfunctions and fatigue failures of electronic equipment may occur at noise levels between 130 and 150 dB. Above 150 dB, it is very likely that malfunctions and fatigue failures will occur—depending upon the length of time the equipment is exposed to these high noise levels.

The effects on personnel of noise levels below about 105 dB are difficult to evaluate, since they are generally related to the mental attitude of the individual and to his previous exposure to various noises. There is little evidence to indicate that the quality of human performance suffers in a noisy environment except where the task involves reaction to auditory stimuli and these are masked by the noise or where the noise provides a source of distraction. The intensity of the distraction can be evaluated in terms of the annoyance produced. Motivation, adaptation, and habituation play large parts in lowering the annoyance level of noise. Prolonged exposure (several hours) to sound intensities of 90 to 100 dB results in a temporary loss of auditory acuity. At higher intensities, permanent cumulative hearing loss may occur. Damage to hearing depends upon the sound intensity, frequency, and duration. The greatest damage to hearing occurs in the 500 to 2,000 cps range. Sudden damage may result from the noise of a blast or explosion. Gradual damage may result from the continual exposure to noise over a long period of time. A steady noise, such as experienced inside a tracked vehicle when underway, is less damaging to hearing than an impulsive noise, such as from a heavy machine gun.

Table 3-5 gives maximum allowable noise levels recommended for the interior of personnel compartments of military wheeled vehicles. Table 3-6 shows noise level criteria established by the U. S. Air Force as the maximum tolerable at a human operator's position and makes allowances for the duration of the exposure. The

TABLE 3-5 MAXIMUM ALLOWABLE NOISE LEVEL FOR PERSONNEL COMPARTMENTS OF WHEELED VEHICLES[16]

Frequency Band, cps	*Sound Intensity,* dB*
< 75	95
75 – 150	90
150 – 300	85
300 – 600	70
600 – 1200	60
1200 – 2400	50
2400 – 4800	40
> 4800	30

*Decibel reference 0.0002 dyne/cm^2.

TABLE 3-6 USAF NOISE LEVEL CRITERIA AT OPERATOR'S POSITION[17]

Frequency Band, cps	*Damage Risk (Less than one hr),* dB*	*Damage Risk 1 to 8 hr,* dB*	*Communication,* dB*
37 — 75	115	106	106
75 — 150	105	96	96
150 — 300	97	88	88
300 — 600	94	85	81
600 — 1200	93	84	75
1200 — 2400	92	83	72
2400 — 4800	91	82	71
4800 — 9600	90	81	69

* Decibel reference 0.0002 dyne/cm^2

noise levels given in the two "Damage Risk" columns are the maximums to which unprotected personnel can be exposed for the periods indicated without risk of permanent hearing damage. The values in the "Communications" column are applicable to situations in which minimum voice communication is required. The noise levels indicated will permit shouted voice communication at distances below about 3 ft. Where frequent or continuous person-to-person, nonelectrically-aided, voice communication is required; the steady state noise levels should not exceed those given in Table 3-7. For more information on maximum allowable noise levels in military vehicles, see latest issue of HEL Standard S-1-63 which can be obtained from the Human Engineering Laboratories, Aberdeen Proving Ground, Maryland.

At sound intensities above 120 dB, serious physiological effects are encountered, particularly under prolonged exposure. These are shown in Table 3-8. Sound intensities of this magnitude are encountered in close proximity to heavy artillery fire and jet engines.

3-2.3 NOISE CONTROL

The noise level in a vehicle can be affected by attacking it at its source, by reducing the effectiveness of the sound transmission paths, or by applying quieting techniques at the listening positions. Attacking the noise problem at its source involves basic design changes to the component or equipment which are the primary sources of the vibration in an effort to eliminate the vibration, reduce it, or alter it to a less objectionable situation. This is the most effective method of noise control, but it is not always feasible or possible to apply. When

TABLE 3-7 MAXIMUM STEADY STATE NOISE LEVEL FOR NONELECTRICALLY-AIDED PERSON-TO-PERSON COMMUNICATION

Frequency, cps	*Noise Level,* dB*
37 — 75	79
75 — 150	73
150 — 300	68
300 — 600	64
600 — 1200	62
1200 — 2400	60
2400 — 4800	58
4800 — 9600	57

*Decibel reference 0.0002 dyne/cm^2

TABLE 3-8 PHYSIOLOGICAL EFFECT OF NOISE AT HIGH INTENSITIES[18]

Frequency, cps	*Intensity,* dB	*Effects*
100 to 12,000	130	Interference with voice communications, permanent cumulative hearing loss.
1,000	135	Inner ear effects, nausea, vomiting, nystagmus, shifting of visual field.
100 to 12,000	140	Auditory pain, permanent cumulative hearing loss even with best protective devices.
100, to 12,000	150	Massive stimulation of many senses, nausea, vomiting, intense ear discomfort even with best protective devices.
100 to 12,000	150 to 160	Severe breakdown of psychomotor performance.
20,000	160	Unusual fatigue, unbearable pains in palms of hands, body heating.

alteration of the primary source of the vibration is not possible, steps should be taken to isolate it from the body and hull. This involves mounting the offending equipment upon suitable vibration isolators that prevent or alter the transmission of undesirable vibrations to parts of the vehicle that may be more effective radiators of sound (such as thin wall panels), causing them to vibrate. Since various equipment whose normal operation involves repeated impacts or sudden accelerations—such as the suspension components, internal combustion engines, rapid fire weapons, and vehicle-mounted artillery—are inherent to military vehicles, as much of these forces as possible should be absorbed within the structure to prevent them from reaching the efficient sound radiating surfaces (wall, floor, or ceiling panels).

Vibrations and shocks, and hence noises, are transmitted from their sources, through structural components and through the junctures between the structural components. The route that a particular vibration takes is known as a noise train. A typical noise train is one that begins with a vertically oscillating wheel, travels through the suspension system, is transferred to the frame, and then travels to the vehicle body or to the cab. Another typical noise train runs from the engine, through the engine mounts to the frame, and then to the body. Within the body, a noise train may extend from the frame to the floor, to the walls, to the roof. At each component juncture there exists an opportunity to absorb or alter the vibration by interposing vibration inhibiting materials between the joints during assembly. Typical materials used for this purpose are rubber mastics and layers of elastomeric materials. This technique interrupts and reduces the effectiveness of the noise train.

Thin panels that have large unsupported areas are prone to vibrate transversely in the fashion

of a large drum head. This type of vibration can often be controlled by providing the panel with additional transverse stiffness in the form of corrugations in the panel or by adding stiffeners. When this technique is not feasible, viscous damping materials can be applied to the vibrating surface. Various nonhardening plastic mixtures such as putty, asphalt, tar, and various elastomeric mastics are used for this purpose. These may be buttered directly onto the panel surface, or they may be used as loading for cloth or felt to facilitate handling and application and to avoid subsequent flow after application. The weight of the damping layer required depends upon the weight and size of the vibrating panel. Relatively rigid coatings are required for stiff panels having high resonant frequencies, and thicker and softer coatings are required for suppresion of low-frequency vibrations.

Airborne noise is transmitted through thin wall panels and partitions principally by minute flexure of the wall as a whole in response to incident sound pressure on the noisy side. This induced vibration results in a reradiation of the sound on the quiet side of the wall. The application of viscous damping materials to these surfaces is also effective in reducing this type of noise transmission. Other effective techniques involve increasing the mass per unit area of the partition, constructing the partition of material that has a high viscosity for bending or by the use of double partitions that are vibrationally isolated from each other.

Once a source of noise is established in a room or compartment, the sound level developed is considerably higher than the same noise source would produce outside of the compartment. This is due to the successive reflections of the sound from the walls, floor, and ceiling. Quieting techniques are employed to prevent this increase in noise level. These involve the application of sound-absorbing materials to all interior reflecting surfaces that are exposed to the noise. The materials used for this purpose derive their sound absorbing properties either from the porosity of their surfaces, which traps and absorbs the sound, or from the dissipative vibration of their surface layers. The effectiveness of these absorbents varies with the frequency of the sound and is usually greater for the high and intermediate frequencies than for low frequencies. It is measured by determining the *absorption coefficient* of the material, which is defined as the fraction of the total sound energy incident on the material that is not reflected. For ordinary noise quieting, another unit is commonly used called the *noise-reduction coefficient.* This is the average of absorption coefficients measured at frequencies of 250, 500, 1,000, and 2,000 cps. Typical values of these coefficients for some representative materials are given in Table 3-9. Absorption coefficients of a large variety of materials are available from the Acoustical Materials Association[19]. The measured absorption coefficient is not a property of the material alone, but is influenced by the size and mounting of the test sample and by the size and shape of the test chamber. Therefore, when comparing the absorption coefficients of different materials as published by various sources, the conditions of the test should be noted.

Noise reduction can be estimated quantitatively for a room or compartment by computing the total sound absorption within the compartment after it has been given a noise-reduction treatment and comparing it with the total sound absorption present before the treatment. The total sound absorption present $\overline{C}$ may be computed by multiplying the noise-reduction coefficient η of each material present in the room by its total exposed area A and summing the resulting products, thus

$$\overline{C} = \eta_1 A_1 + \eta_2 A_2 + \eta_3 A_3 + \ldots + \eta_n A_n \quad (3\text{-}12)$$

$$= \sum_{i=n}^{n} \eta_i A_i$$

If $\overline{C}_1$ and $\overline{C}_2$ are the total sound absorptions of the compartment before and after being given a noise-reduction treatment, the noise reduction is evaluated as follows:

$$\text{Noise reduction, dB} = 10 \log \frac{\overline{C}_2}{\overline{C}_1} \quad (3\text{-}13)$$

Greater accuracy in calculating total absorptions is obtained when the frequency spectrum of the noise is known. In these cases, replace the noise-reduction coefficient in Eq. 3-12 with the absorption coefficient of the material measured at the frequency of maximum loudness level.

TABLE 3-9 SOUND ABSORPTION COEFFICIENTS OF SELECTED MATERIALS[19,24]

Materials	Absorption Coefficients						Noise Reduc-tion	Weight, lb/ft²	Surface Description	AMA* Test No.
	Frequency, cps									
	128	256	512	1,024	2,048	4,096				
Acoustical Materials										
Acousti-Celotex**										
Type C-9, 3/4 in.	0.11	0.23	0.80	0.93	0.58	0.50	0.65	0.96	Multiperforated & painted	46-132
Type M-1, 5/8 in.	0.07	0.21	0.64	0.86	0.93	0.83	0.65	1.31	Multiperforated & painted	46-12
Acoustifibre, 5/8 in.+	0.10	0.16	0.62	0.97	0.81	0.73	0.65	0.56	Multiperforated & painted	46-137
Acoustone F, 11/16 in.±	0.08	0.25	0.76	0.84	0.78	0.73	0.65	1.35	Fissured & painted	46-50
Airacoustic, 1 in.°	0.29	0.31	0.70	0.82	0.79	0.80	0.70	1.50	Unpainted	46-71
Cushiontone A, 3/4 in.°°	0.10	0.28	0.66	0.91	0.82	0.69	0.65	1.05	Multiperforated & painted	47-28
Fiberglas Acoustical Tile plain, 3/4 in.‡	0.04	0.20	0.63	0.91	0.82	0.82	0.65	0.69	Painted	A48-99
Fibretone, 13/16 in.°	0.14	0.37	0.69	0.80	0.76	0.73	0.65	1.17		46-124
Q-T Ductliner, 1 in.**	0.21	0.42	0.71	0.86	0.79	0.75	0.70	1.3	Unpainted	A48-10
Sanacoustic, KK pad plus metal facing & pad supports, 1-9/16 in.°	0.25	0.58	0.96	0.97	0.85	0.72	0.85	pad only, 1.28	Multiperforated, enameled metal pan backed with wool pad	46-88
Travertone, 3/4 in.	0.06	0.23	0.78	0.97	0.84	0.80	0.70	1.20	Fissured & painted	
Brick wall	0.012	–	0.017	–	0.023	–	0.02		Painted	
Concrete wall	0.01	–	0.015	–	0.02	–	0.02		Smooth	
Cork or rubber tile on concrete	–	–	0.03 to 0.08	–	–	–	0.05			
Glass	0.035	–	0.027	–	0.02	–	0.02			
Metal, heavy section	0.01	–	0.013	–	0.015	–	0.02			
Wood	0.05	–	0.03	–	0.03	–	0.03			

* Acoustical Materials Corp.
** The Celotex Corp.
+ National Gypsum Co.
± United States Gypsum Co.
° Johns-Manville Corp.
°° Armstrong Cork Co.
‡ Owens-Corning Fiberglas Corp.

In general, the larger the area of sound absorbing material and the higher its noise-reduction coefficient the more effectively will the noise be reduced. Thus, quieting techniques are more effective in large rooms, rooms that are large with respect to the wavelengths of the sounds involved, than they are in small rooms. This precludes the application of quieting techniques to small crew compartments of military vehicles—or it makes such application very difficult at best. The most effective and practical means of noise control in a vehicle is by controlling noise at its source and by applying isolating and damping techniques. Considerable information is available in the open literature on acoustics and noise control. Refs. 20 to 25 are particularly good.

SECTION II—DOORS, GATES, GRILLES, AND ACCESS OPENINGS

Vehicle bodies and hulls require adequate means of access to their interiors. These are necessary for the accomplishment of such functional objectives as personnel entrance and egress; loading and unloading of cargo; equipment servicing, maintenance, inspection, and replacement; ventilation of crew, passenger, and fighting compartments; and for the provision of air for power plant operation and cooling. In most instances, these openings require closures for a variety of reasons—such as for protection from inclement weather, for ballistic protection, for protection from CBR agents and dust, for the retention of cargo, and to maintain the structural integrity of the body or hull. The last is especially important in the case of amphibious vehicles. These closures take the form of doors, gates, grilles, and covers. Each of these is discussed separately in the paragraphs that follow. First, however, some general guidelines are given which apply to all types of access openings and closures:

a. Access openings should be left without closures in all instances where it is practical to do so.

b. All openings should be sufficiently large to permit the required activity to be performed by personnel wearing arctic clothing and, where required, to permit adequate visual observation of any equipment or components that require manipulation.

c. All closures that are not completely removable should be made self-supporting in their open positions, unless the design is such that they will remain in their open positions due to the action of gravity.

d. To the extent possible, all doors and covers should be designed in a way that will make their method of opening obvious to the user. In cases where this is not possible, a clearly worded instruction plate should be permanently attached to the outside of the closure.

e. It should be readily apparent when a closure is in place but is not properly secured.

f. Avoid all sharp edges and corners on all types of access openings and on their closures.

g. The opening or removal of closures should not be prevented by any obstruction on the vehicle, such as the turret, vehicle-mounted weapon, on-vehicle equipment, or by structural members.

h. When a hinged closure is used, a clearance space should be provided equal to the volume swept by the cover as it is swung to its fully opened position.

i. Hinged closures should be provided with quick-opening latch locks or with quick opening captive fasteners.

j. Use a minimum number of fasteners to retain a closure, consistent with flotation, ballistic, CBR, and safety requirements.

k. Maximum use should be made of tongue-and-slot catches to minimize the number of fasteners required.

l. All fasteners should be of the captive type and, where possible, identical.

3-3 DOORS AND HATCHES

3-3.1 GENERAL DISCUSSION

A door is any movable piece of firm material, such as metal, wood, or plastic—or it can consist of a structure comprising several parts—by means of which an opening to an enclosure may be closed or opened for access or exit. It is distinguished from a cover or cover plate in that a door is permanently attached to the enclosure, usually along one side or along the top or bottom edges, by means of hinges or pivots

about which it can swing to open or close the opening. This is the type of door most commonly found on military vehicles. Not all doors are hinged, however. In some configurations, the doors are retained in grooves or ways along which they are made to slide; in others they are suspended from a trolley arrangement along which they roll. Some doors are made to fold up like an accordion, and still others are fabricated of narrow strips which permit the door to be rolled up and down or to one side. All doors, however, remain attached to the enclosure during their opening and closing operations, and their removal requires partial disassembly of the door retaining structure.

The term *hatch* is frequently used with reference to the closure of an access opening, particularly one in the hull of a ship, aircraft, or hull-type military vehicle. It is specifically applicable to a door or removable cover that gives vertical access into a compartment. Thus, the access doors in the top of a tank turret, or the driver's access door in the top of the tank hull, are properly called hatches.

Doors and hatches can be armored or unarmored, load bearing or load free, of solid or open construction. The type of door used and its construction depend upon the mission of the vehicle and the hazards to which the door will be exposed. The normal procedure is to construct the door of the same material as is used for the body or hull in the immediate vicinity of the door opening. Proper framing and reinforcing are required around the door opening to maintain necessary rigidity. Armored doors are cast, forged, or of welded construction. Doors for unarmored vehicles are welded of solid plates or are fabricated of sheet metal using a hollow core-type construction. Fig. 3-3 shows an exploded view of the driver's cupola cover, or hatch, used on the M107, 175 mm Self-propelled Gun that is illustrated in Fig. 1-30. This cover design uses a torsion bar hinge-pin to counterbalance the weight of the cover. Fig. 3-4 is a top view of the M60 Tank upon which many different types of hatches and covers are visible.

3-3.2 DESIGN REQUIREMENTS

Whenever it is practical to do so, access openings should be left open and not be provided with a closure. Only in cases where closures are necessary should they be provided. The paragraphs which follow give general guidance in the design of various types of doors and hatches.

Cargo doors or hatches should be as large as possible, consistent with the type of cargo to be stowed and the hull integrity requirements. They should be strong enough to withstand surf and water loads, specified ballistic impacts, normal cargo handling impact loads, and general handling and transportation loads. They should be positioned so as to provide the easiest access route to the cargo hold, should be easily opened and closed, and they should be provided with a lock to secure the cargo. Normally, cargo doors or hatches are constructed of the same type of material as the adjacent body or hull. Since military cargo vehicles are tactical vehicles, their doors or hatches are generally made of aluminum alloys, steel, or other metals. Proper framing and reinforcing are required to achieve the necessary degree of door rigidity.

Personnel doors and hatches should be positioned for easy entrance; should be large enough to accommodate troops in arctic clothing and full combat gear; be sufficiently rigid to survive specified ballistic impacts; possess strength to resist handling and water loads; and, when necessary, should be sealed to keep out water and to provide an air-tight seal for CBR protection. Personnel doors are generally made of aluminum, mild steel or armor plate, depending upon the door requirements and the primary material of the vehicle. Armored doors require particular design and construction care so that they can give the same level of ballistic protection as the adjacent body or hull structure.

Equipment access doors should be positioned close to the equipment to which they give access and should be large enough to provide ready access for repair and removal. If hinged, a door should either swing into a neutral position where it is retained by normal gravitational forces, or it should be provided with a positive mechanical means of retaining the opened position. Equipment access doors should be provided for every piece of covered equipment that requires periodic inspection, maintenance, or removal; such as engines, transmissions, electronic gear, CBR filters, pumps, and life support equipment. Equipment access doors are generally fabricated of sheet metal with formed sheet metal

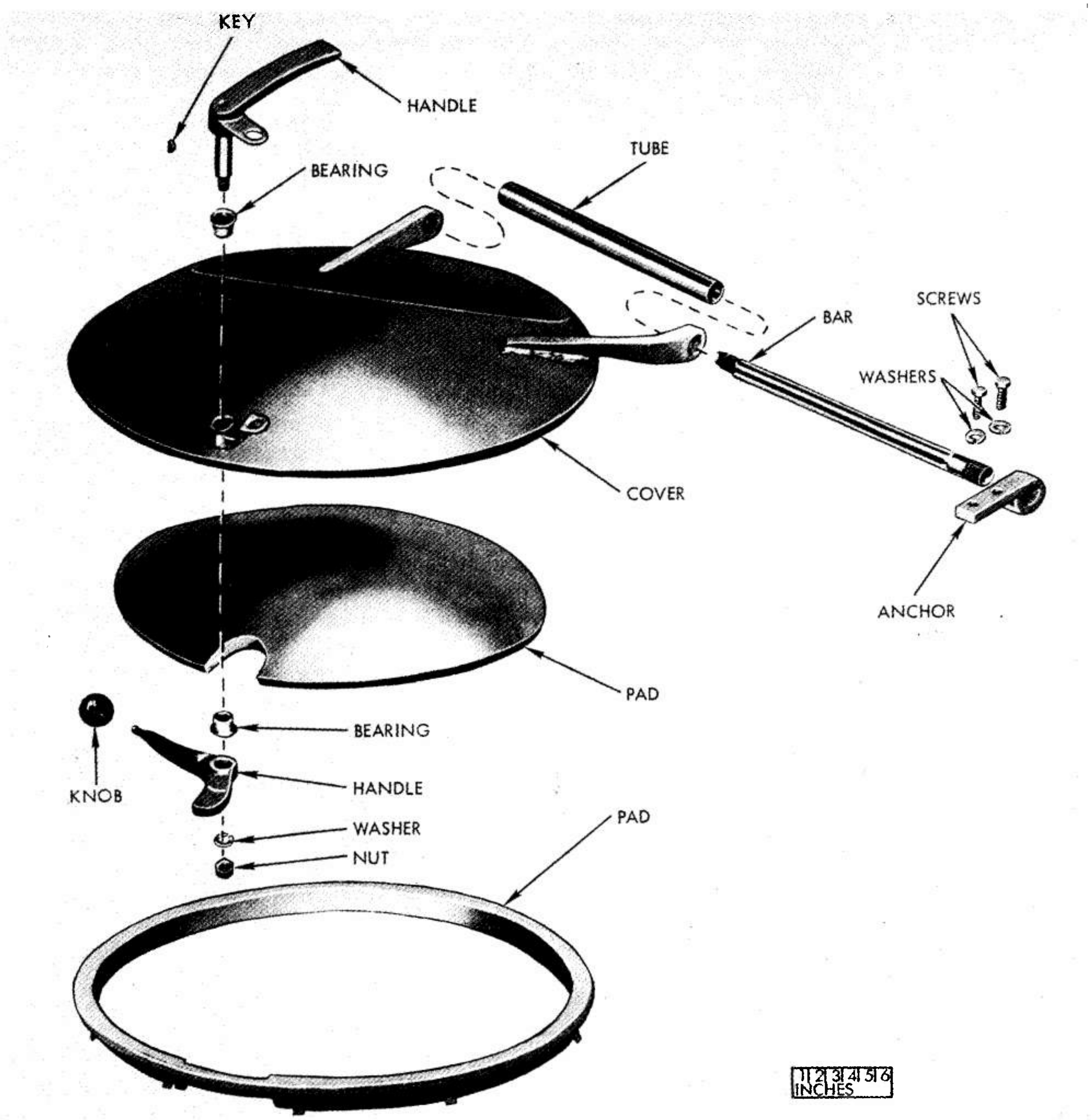


Figure 3-3. Exploded View of Driver's Cupola Cover, M107, 175 mm Self-propelled Gun

stiffeners. Exceptions are equipment access doors on armored vehicles—these are made of armor plate.

Escape hatches should be provided in all closed combat and tactical vehicles. They should be large enough to permit the egress of troops in combat gear; strong enough to give a level of protection equal to the adjoining hatch area; sealed against weather, water, and CBR agents when required; capable of being locked from the inside; and positioned to provide the maximum escape route protection and require the minimum egress time. Combat tanks are usually provided with an escape hatch located in the underside of the vehicle. This exit is of particular valı e in the event the tank is afire, as the heat anı. flames naturally reach upward making escape through the normal hatches

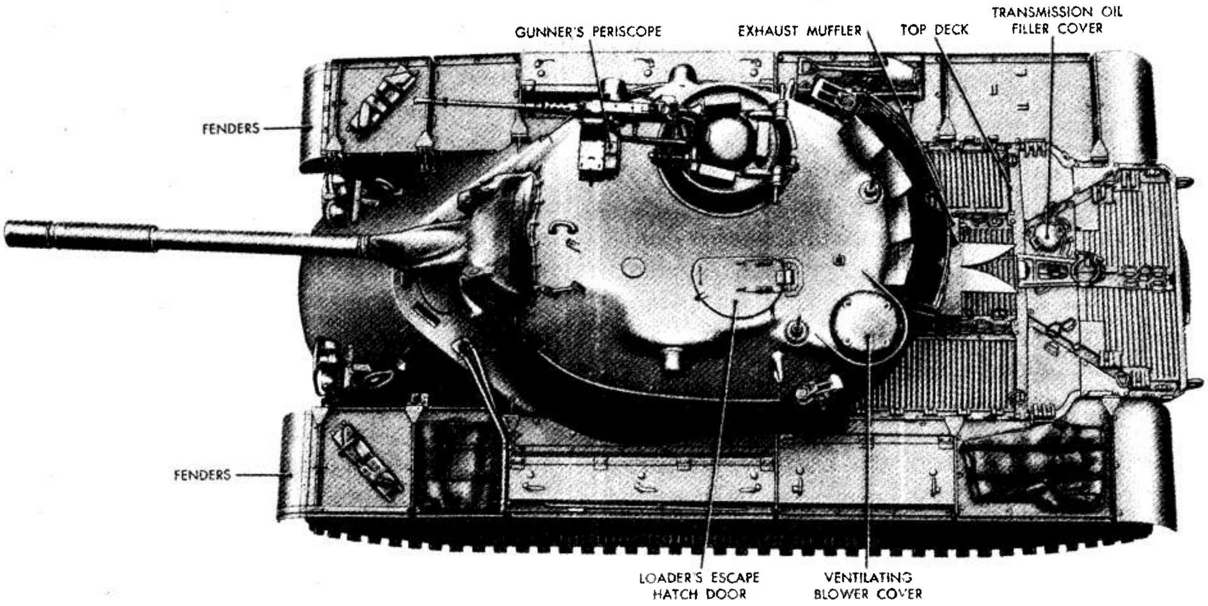


Figure 3-4. Top View of M60 Tank, 105 mm Gun

virtually impossible. Amphibious vehicles are usually left open at the top to facilitate escape during serious emergencies while waterborne. Fully enclosed amphibious vehicles of the M113 type (Fig. 1-25) are generally provided with escape hatches in their top where they can be opened without danger of immediately swamping the vehicle.

A turret or hull door is particularly susceptible to damage from shock because the entire unit and all its working parts can be severely strained with one impact. Doors which are not flush with the surrounding plate must be designed with adequate thickness around their edges to resist penetration. Also, any reduced obliquity of the door dictates a need for greater thickness on the door areas. If possible, door hinges should be made an integral part of the cover and should be attached to the interior side to reduce ballistic damage.

Before World War II, nearly all doors on tanks were vulnerable to bullet splash (par. 2-13.3.4). This weakness was overcome by stressing the two most effective means for defeating splash—bullet deflectors and splash traps (Fig. 3-5). Involute traps direct the splash into an ever-decreasing radius and consequently slow it down rapidly. Wedge-type traps depend on the splash bouncing back and forth in the wedge until it loses energy and becomes ineffective. If armor clearances are desired, three right-angle turns are considered the minimum number

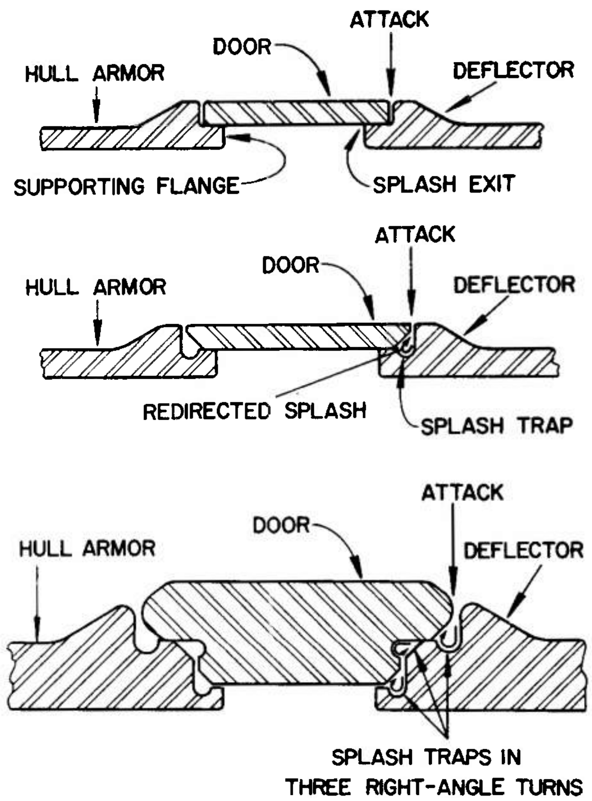


Figure 3-5. Bullet Deflectors and Splash Traps

necessary to expand the harmful energy of splash. For more detailed coverage of this subject, see Ref. 26.

3-4 GATES

Gates are particular types of doors; therefore, most of the general requirements given for door design apply equally to gate design. Typical gates are the sidegates and tailgates on dump, stake, and cargo trucks and trailers. Normally, a gate is distinguished from a door by the manner in which it is hinged. A door is generally hinged at one side while a gate is horizontally mounted and is hinged at the top, the bottom, or both (selective openings). Gates should be designed so they can be positioned and retained in the closed position, in the open position, or in any intermediate position. Normally chains and hooks are provided as tailgate position supports. The hooks are also used to lock the gate in the closed position (Fig. 1-20).

The construction of gates varies. Most gates are fabricated of built-up sections of sheet metal. Others are of a sandwich structure which may have a honeycomb core. The gate structure is generally similar to the structure of the vehicle wall, since it is subjected to many of the same loads. Occasionally, a gate is used as a ramp or as a platform (by being placed in a horizontal position). In these applications, the gate may be subjected to suddenly applied concentrated loads which have to be borne by the gate structure. Therefore, hinges and points of support must be designed to withstand vertical as well as horizontal loads. Support points must also be capable of distributing the loads evenly to the gate structure.

3-5 GRILLES

3-5.1 TYPES

Air is required in the interior of vehicles for engine operation and cooling and for ventilation of the crew and fighting compartments. Openings in the body or hull for the admission and discharge of air must be protected from the entry of large foreign objects such as rocks, branches, miscellaneous debris and, in the case of armored vehicles, from ballistic matter such as bullets and projectile fragments. This protection is afforded by a variety of grilles placed over the openings. An assortment of grille designs is used including latticed, vaned, louvered, and baffled structures made of aluminum, mild steel, or armor plate.

Tactical vehicles generally utilize simple latticed grilles for radiator covers, air intakes, and exhaust ports. Armored vehicles generally use special armored grilles designed to deflect the larger ballistic objects and trap the smaller ones. Amphibious vehicles have a special grilles problem, that of providing passage for a sufficient quantity of air without allowing an excessive amount of water to enter. They utilize louvered and baffled grilles for this purpose. These louvers and baffles are sometimes made movable for selective positioning during land or waterborne operations. Some amphibious vehicles have water-sensitive grilles that close momentarily when a wave breaks over them and open again when the water recedes. Adjustable louvered grilles are also used on certain lightly armored vehicles; however, the ballistic impact energy that can be absorbed by these grilles is fairly low which makes other types of armored grilles more popular.

3-5.2 GENERAL REQUIREMENTS

The air intake and exhaust openings of a vehicle are usually placed in an area which is as free from dust and dirt as possible and which provides the closest practicable location to the point of air utilization. For this reason, air intakes for ventilation purposes are located in the upper portion of the vehicle, while engine air intakes and exhausts are located in the front or rear portion of the vehicle, depending upon the engine location.

Grille designs for open tactical land vehicles are relatively simple owing to their usage. The open lattice construction of these grilles offers no appreciable restriction to air flow; therefore, the designer need only cover the opening with a grille that is strong enough to resist the impacts of branches and other relatively large objects. Armored combat vehicles and amphibians, which require a partially closed grille, offer a design challenge. For these vehicles, the grille must be optimally located, be of minimum size to reduce the amount of foreign matter which must be trapped or deflected, and must provide sufficient air flow. In armored vehicles particularly, the positions of the grille are

compromised, since some positions which offer an optimum air flow may be most vulnerable to ballistic attack. Similarly, in amphibious vehicles the wave splash locations may dictate the relocation of grilles to drier vehicle areas.

In the efforts to meet the diametrically opposed requirements of providing passage for a sufficient volume of air to satisfy the operating and cooling requirements of the power plant and at the same time provide maximum protection for the power plant against projectiles and fragments, various arrangements and cross-sectional contours of parallel bars have been tried. Some offer a compromise between the air flow desires and the ballistic protection requirements. One of the most efficient contours from both the standpoint of protection and air flow is the modified single-trap grille illustrated in Fig. 3-6(F). An asphalt-like adhesive material, indicated by the crosshatched area, may be added to the trap area. The adhesive forms a smooth airflow surface, catches some of the low-velocity fragments and bullet splash, and is soft enough not to interfere with the bullet and fragment-trapping capabilities of the grille.

The tabletop grille illustrated in Fig. 3-7 is another means of effecting the airflow-protection compromise. It consists of a solid protective cover which is suspended to

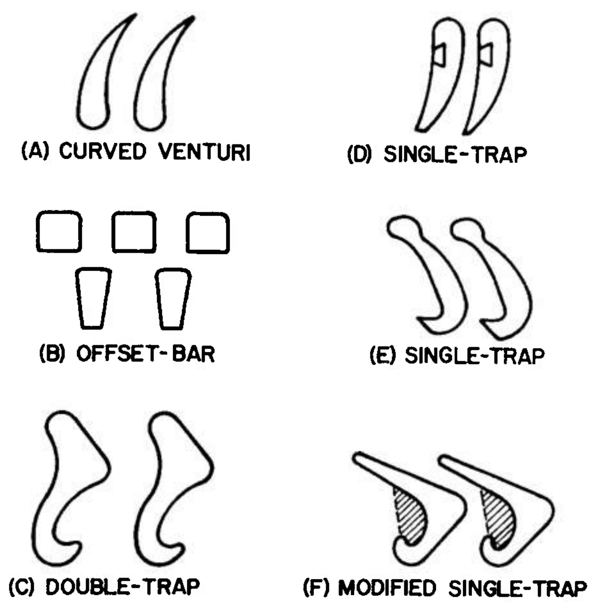


Figure 3-6. Cross Sections of Various Grille Configurations

allow air to pass into the engine compartments. With the addition of baffles to absorb fragments and wire mesh to prevent hand grenade entry, this type of grille provides very good protection against ballistic attack. However, its inherent air restriction is high, limiting its application to low volume, high pressure cooling systems. Guide vanes may be employed to improve the air flow characteristics of the tabletop grille; however, design problems are involved in their construction. The vanes must be sufficiently rigid to withstand vibrations imposed by the engine, and yet must not be so strong that they act as splash guides for ballistic fragments into the engine compartment. For additional information on this subject see Ref. 26.

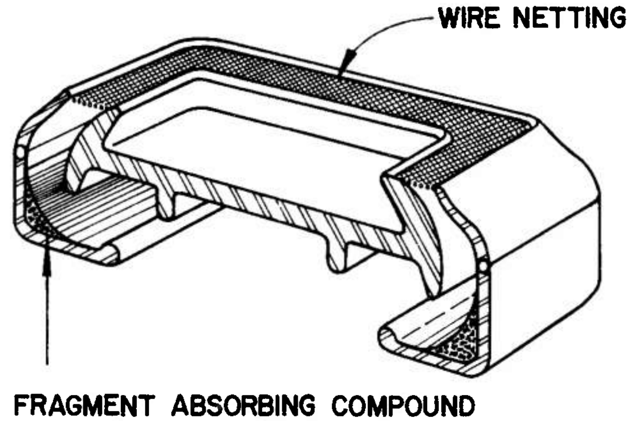


Figure 3-7. Cross Section of Tabletop Grille

3-6 ACCESS OPENINGS

3-6.1 TYPES

During the normal vehicle life, portions of the vehicle require inspection, servicing, adjusting, maintenance, repair, removal, or replacement. In order to facilitate these functions, openings are required in the vehicle body or hull to provide access to all components that are not accessible from the inside or components that specifically require access from the outside. Typical access openings are those for the engine and transmission, brake linkage, hydraulic drains, filters, air cleaners, radio or electronic gear, generators, starters, and special equipment. The openings vary from small ones with transparent covers to allow visual inspection to large ones providing access for the removal of the entire power plant.

The access opening construction varies with the type of vehicle and the position of the opening on the vehicle. Interior access openings and those on tactical vehicles can be of relatively simple construction. Some are no more than holes cut into a panel; others may require some beading to achieve the required rigidity; still others may require stiffening and strengthening by the addition of metal bracing and framing. On amphibious and armored vehicles, some access openings may be in a position in which water pressure or ballistic loads may be expected. In these cases, the access opening designs may require extensive reinforcing and provisions for sealing so that the access closures may be properly supported and sealed.

In general, the type, size, shape, and location of access openings should be based upon a thorough consideration of the following factors:

a. Operational location, setting, and environment of the unit being provided with access

b. Frequency of use anticipated for the access

c. Tasks to be performed through the access

d. Time required to perform the above tasks

e. Types of tools and accessories that will be required in the performance of above tasks

f. Work clearances required

g. One hand or two hands required to perform tasks

h. Type of clothing the technicians are likely to be wearing

i. Distance into access that technicians must reach (finger, hand, arm, more)

j. Visual requirements of task

k. Packaging of subassemblies and components that are to be serviced behind the access

l. Mounting of subassemblies and components behind the access

m. Hazards in using the access

n. Size, shape, weight, and clearance requirements for logical combinations of human appendages, tools, equipment, removed or replacement items, etc., that must pass through the access

Functionally, access openings are of three general types (1) visual inspection ports, (2) test and servicing ports, and (3) physical access ports.

Visual inspection ports. These are used where only visual access is required to inspect the condition of equipment, read a gage, etc. For this purpose, it is most desirable to use no cover whatever, unless exposure to the surrounding environment is likely to degrade the equipment or performance of the system. In those cases where dirt, moisture, or other foreign materials are a problem, a transparent plastic window should be used to cover the opening. In situations where physical wear, contact with solvents, or other factors would cause optical deterioration of the plastic, substitute a break-resistant glass for the plastic. Where the cover will be subjected to high stresses, pressure, or where considerations of safety preclude the use of glass; provide a metal cover plate fitted with an opening device, where practical, or retain the cover plate with the smallest number of the largest screw fasteners that otherwise suit the requirements.

Test and servicing ports. These give access to test points and to equipment that require frequent servicing such as filters, air cleaners, lubricating systems, etc. Here, too, it is preferable to have no cover, unless exposure is likely to degrade equipment or system performance. In situations where the entrance of dirt, moisture, or foreign material would create an unacceptable condition, spring-loaded hinged or sliding closures should be used. Where this type of closure will not satisfy stress, sealing, or other requirements, use a cover plate retained by quick-opening fasteners.

Physical access ports. These are used where the maintenance, repair, or replacement task requires that large portions of the body (hands, arms, head, and shoulders, etc.) be inserted through the access opening and that subassemblies, components, tools, and equipment be passed through it. In these cases, if the system cannot tolerate exposure to dirt, moisture, or foreign objects, use a hinged door for closure and hold it in place by screws or other fasteners. Where lack of space precludes the use of a hinged closure, use a cover plate retained by captive screws or other captive, quick opening fasteners. Where this is not feasible due to stress, pressure, safety, or other requirements, use a properly designed closure retained by smallest number of the largest screw fasteners suitable to the design. Fig. 3-8 shows various types of access plates, covers, and doors on the bottom of an armored hull.

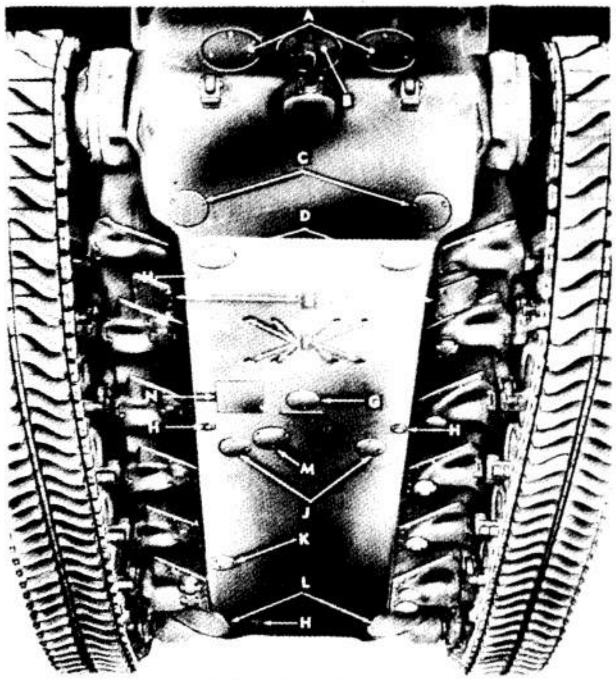


Figure 3-8. Bottom View of Armored Hull Showing Various Access Plates, Covers, and Doors

3-6.2 GENERAL REQUIREMENTS

Since the end of World War II, military vehicle design has changed drastically. Modern weapon technology, coupled with the desire for more off-road vehicle mobility, has resulted in the design of tactical and combat vehicles that are more enclosed than their predecessors, which enables them to cross inland waterways; they are of lighter weight, to increase their maneuverability; and they are generally more complex. For the body and hull designer, the changes have brought about revisions in the types of materials used (more aluminum, titanium, and magnesium) and increases in the stress levels carried by the body or hull structure.

As the vehicle becomes more complex, accessibility becomes more important and, correspondingly, more difficult to achieve in a closed, highly stressed body than in an open-type vehicle. Access openings, therefore, must not only be judiciously placed to insure the proper degree of accessibility, but they must also be reinforced to retain body or hull strength and to effect efficient seals when required.

Access must be provided not only for the normal repair of all major vehicle components, but also for the repair of components damaged by enemy action. All combat and tactical vehicles have a probability of being damaged by battle or sabotage. While the type of damage inflicted may be of a random nature, the vehicle designer must design ready access to all components for which battle damage would not result in a total vehicle loss.

Riveted panels and cover plates are never to be used as access points. The overall layout and design of body or hull structures should never require the removal of permanently attached structures, not even for infrequent maintenance.

Accesses need not have regular geometric shapes; in fact, nonsymmetrical shapes can be used to advantage in assuring that closures will be replaced the same way each time they are removed—when this is a requirement. Similarly, different shaped openings prevent the erroneous interchange of closures in cases where the closures are completely removable. When nonsymmetrical closures are used as a code to correct replacement, make the lack of symmetry obvious to facilitate assembly. Asymmetrical spacing of mounting holes can also be used for this purpose.

The size and shape of an access should be such as to permit an easy passage of the items involved, including the technician's hands, arms, or body, tools, equipment, clothing, etc. Toward this end, the following points should be considered:

a. Dimensions of the various items that must be replaced through the access

b. All protuberances, attachments, handles, etc., on the items

c. Methods of grasping the items during removal and the clearances required

d. Clearance required within the compartment to perform the tasks necessary

e. The technician's need to see what he is doing inside the compartment

f. Maximum weight that can be handled safely and effectively in body position necessary to accomplish required task. In this respect, the maximum weight that can be held in the hands,

when the arms are fully extended and the body is in a position other than standing, is 25 lb.

In general, one large access is better than two or more small ones; visual and physical access may be provided separately when structural or other considerations require it. Tables 3-10, 3-11, and 3-12 give some quantitative guidance relative to the size requirements of access openings for various tasks. Additional information of this type is given in Refs. 16 and 27-30.

SECTION III—SEALS

3-7 PURPOSE

The function of seals is to prevent or minimize the passage of liquids, gases, particulate matter, or even light across openings and joints between separable members of mechanical and structural assemblies. In military vehicles, they are used to prevent the leakage of lubricants, fuels, coolants, and exhaust gases from their containing vessels; to prevent the entrance of contaminants such as dust, dirt, sand, mud, snow, water, air, and chemical warfare agents; and to maintain the buoyancy and water-tightness of amphibious and deep-fording vehicles. Since military vehicles are required to operate under blackout conditions, an application that is somewhat unique to these vehicles is that of sealing joints and openings in the body or hull against the outward leakage of light. Generally, a seal that is effective against liquids, gases, and fine dust particles is also an adequate light seal; but the converse does not necessarily apply.

The term "seal" is usually interpreted as a general name for all means of preventing the migration of matter (and light) across a joint or opening. Thus, it includes all types of packings and gaskets. The interpretation generally given to these last two terms is as follows. A *packing* is a dynamic seal used where there is some relative motion between rigid members, as between a reciprocating piston rod and the cylinder bearing or between the propeller drive shaft and the hull of an amphibious vehicle. A *gasket*, on the other hand, is a seal used where there is no relative motion between the joined parts, as between a door and its frame or between a removable access plate and the frame of the access opening.

The military environment presents sealing problems that are unmatched by any other environment. Military vehicles are expected to perform in environments from which their civilian counterparts normally turn back; as a result, military vehicles are subjected to many times more dirt, mud, sand, snow, water, heat, and chemical agents than are civilian vehicles. Seals that are adequate in civilian vehicles often deteriorate, leak, or wear out quickly when exposed to the military environment. The prinicipal factors that lead to deterioration or failure of seals in military vehicles are sand, mud, oils, water, heat, extreme cold, and ozone. In addition, amphibious and deep-fording operations (see par. 2-2.1.6) require that seals on military vehicles maintain watertight compartments when subjected to an appreciable hydraulic head.

Many excellent references exist covering the design of seals and joints and on the characteristics and applications of various sealing materials. For a more comprehensive treatment of this subject consult Refs. 5 and 31-40.

3-8 SEAL TYPES AND MATERIALS

3-8.1 SIMPLE GASKETS

A simple gasket is defined as any sealing device whose function is to create and maintain an impervious barrier against the transfer of fluids, gases, or particulate matter across the interface of separable surfaces of a mechanical assembly which do not move relative to each other. The materials used form the impervious barrier by being compressed under an initial squeeze and filling all irregularities between the compressing surfaces. Therefore, to be most suitable, gasket materials should be relatively soft and elastic. The materials most commonly used are cork and cork compositions, cork and rubber, asbestos fiber and rubber, cellulose fiber and rubber, rubber alone (natural and synthetic), copper, and lead. Metallic gaskets are used in high pressure applications where the possibility exists of soft materials being extruded out by the pressure.

TABLE 3-10 MINIMUM DIMENSIONS OF ACCESS OPENINGS FOR HANDS AND ARMS

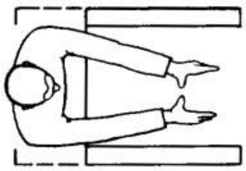
MINIMAL TWO-HAND ACCESS OPENINGS :

REACHING WITH BOTH HANDS TO DEPTH OF 6 TO 25 INCHES:

- LIGHT CLOTHING: 5" HIGH BY 8" OR 3/4 DEPTH OF REACH*
- ARCTIC CLOTHING: 7" HIGH BY 6" PLUS 3/4 DEPTH OF REACH

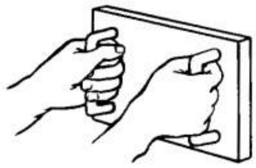
REACHING FULL ARM'S LENGTH (TO SHOULDERS) WITH BOTH ARMS:

- WIDTH - 19-1/2 INCHES, HEIGHT - 4 INCHES



INSERTING BOX GRASPED BY HANOLES ON THE FRONT:

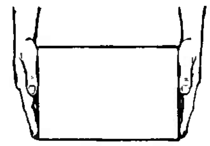
- 1/2" CLEARANCE AROUND BOX, ASSUMING ADEQUATE CLEARANCE AROUND HANDLES



INSERTING BDX WITH HANDS DN THE SIDES:

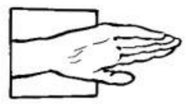
LIGHT CLDTHING:	WIDTH:	BOX PLUS 4-1/2"
	HEIGHT:	5" OR 1/2" AROUND BOX*
ARCTIC CLOTHING:	WIDTH:	BOX PLUS 7"
	HEIGHT:	8.5" OR 1/2" AROUND BDX*

NDTE: IF HANDS WILL CURL AROUND BOTTOM DF BOX, ALLOW AN AODITIONAL 1-1/2" IN HEIGHT FDR LIGHT CLOTHING. 3" FOR ARCTIC CLOTHING.

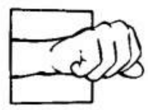


MINIMAL ONE-HAND ACCESS OPENINGS :

	WIDTH	HEIGHT
EMPTY HAND TO WRIST:		
BARE HAND, ROLLED:	3.75" SQ. OR DIA.	
BARE HAND, FLAT:	2.25"	X 4.0" OR 4.0" DIA.
GLOVE OR MITTEN:	4.0"	X 6.0" OR 6.0" DIA.
ARCTIC MITTEN:	5.0"	X 6.5" OR 6.5" DIA.



CLENCHED HAND, TO WRIST:		
BARE HAND:	3.5"	X 5.0" OR 5.0" DIA.
GLOVE OR MITTEN:	4.5"	X 6.0" DR 6.0" DIA.
ARCTIC MITTEN:	7.0"	X 8.5" OR 8.5" DIA.

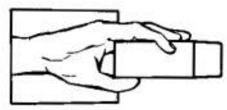


HAND PLUS 1" DIA. DBJECT, TO WRIST:

BARE HAND:	3.75" SQ. OR DIA.
GLOVED HAND:	6.0" SQ. OR DIA.
ARCTIC MITTEN:	7.0" SQ. OR DIA.

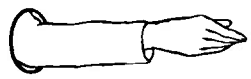
HAND PLUS OBJECT OVER 1" DIA. TO WRIST:

BARE HAND:	1.75" CLEARANCE AROUND DBJECT
GLOVE DR MITTEN:	2.5" CLEARANCE AROUND OBJECT
ARCTIC MITTEN:	3.5" CLEARANCE AROUND OBJECT



ARM TO ELBDW:

LIGHT CLOTHING:	4.D" X 4.5" OR 4.5" DIA.
ARCTIC CLOTHING:	7.0" SQ. OR DIA.
WITH DBJECT:	CLEARANCES AS ABOVE



ARM TO SHDULOER:

LIGHT CLOTHING:	5.D" SQ. OR OIA.
ARCTIC CLOTHING:	8.5" SQ. OR DIA.
WITH DBJECT:	CLEARANCES AS ABOVE

MINIMAL FINGER ACCESS TO FIRST JOINT:

PUSH BUTTDN ACCESS:

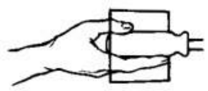
- BARE HAND: 1.25" DIA.
- GLOVED HANO: 1.5" DIA.

TWD FINGER TWIST ACCESS:

- BARE HAND: 2.0" DIA.
- GLOVED HANO: 2.5" DIA.

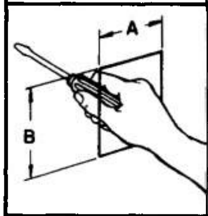
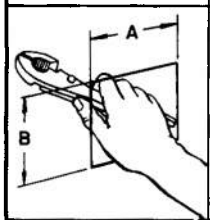
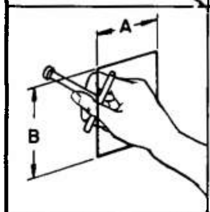
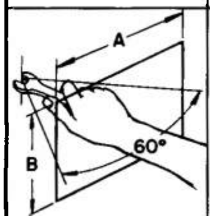
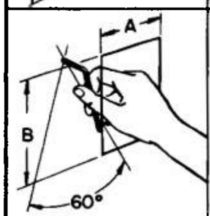
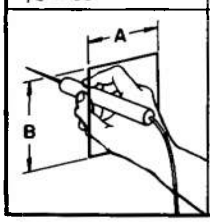
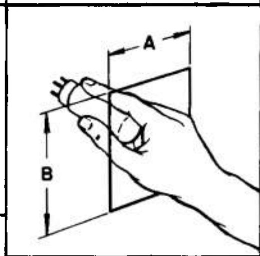
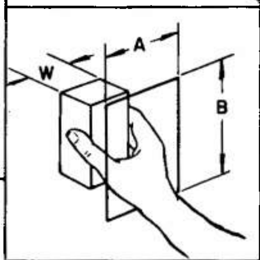
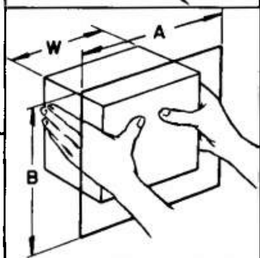
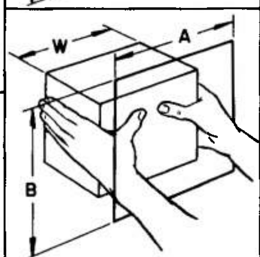
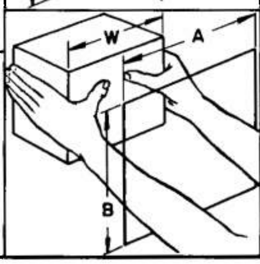
VACUUM TUBE INSERT (TUBE HELD AS AT RIGHT):

- MINIATURE TUBE: 2.0" DIA.
- LARGE TUBE: 4.0" DIA.



* WHICHEVER IS LARGER

TABLE 3-11 DIMENSIONS OF ACCESS OPENINGS FOR ONE- AND TWO-HANDED TASKS[27]

DESCRIPTION OF ACCESS	DIMENSIONS, IN. A	DIMENSIONS, IN. B	DESCRIPTION OF TASK
	4.2	4.6	USING COMMON SCREWDRIVER WITH FREEDOM TO TURN HAND THROUGH 180°
	5.2	4.5	USING PLIERS AND SIMILAR TOOLS
	5.3	6.1	USING "T" HANDLE WRENCH WITH FREEDOM TO TURN HAND THROUGH 180°
	10.6	8.0	USING OPEN-END WRENCH WITH FREEDOM TO TURN HAND THROUGH 60°
	4.8	6.1	USING HEX SOCKET SCR WRENCH WITH FREEDOM TO TURN THROUGH 60°
	3.5	3.5	USING TEST PROBE, ETC.
	4.2	4.7	GRASPING SMALL OBJECTS (UP TO 2-1/16" DIA) WITH ONE HAND
	W+ 1.75	5.0*	GRASPING LARGE OBJECTS (2-1/16" OR MORE WIDE) WITH ONE HAND
	W+ 3.0	5.0*	GRASPING LARGE OBJECTS WITH TWO HANDS; HANDS EXTENDED THROUGH OPENINGS TO BASE OF FINGERS
	W+ 6.0	5.0*	GRASPING LARGE OBJECTS WITH TWO HANDS; ARMS EXTENDED THROUGH OPENING UP TO WRISTS
	W+ 6.0	5.0*	GRASPING LARGE OBJECTS WITH TWO HANDS; ARMS EXTENDED THROUGH OPENING UP TO ELBOWS

* OR SUFFICIENT TO CLEAR PART IF PART IS LARGER THAN 5.0"

TABLE 3-12 MINIMUM DIMENSIONS OF ACCESS OPENINGS FOR UNUSUAL WORK POSITIONS[16,27-29]

Access or Task	*Dimensions*, in.	
	Ordinary Clothing	*Arctic Clothing*
Insert head	8.0 X 10	12 sq or dia
Pass shoulder width	20	27
Pass through prone	20 X 17	27 X 24
Pass through on hands and knees	20 X 31	27 X 38

The selection of gasket material is a function of the temperature and pressure of the sealed fluid. To determine whether a metallic or nonmetallic gasket material should be used, a rule of thumb is to multiply the pressure in psi by the temperature in °F. If the product exceeds 250,000, metallic gaskets should be used. In addition, metallic gaskets should be used in all applications where the temperature exceeds 850°F or where the pressure exceeds 1200 psi.

When designing pressure sealing members using gaskets, the following guidelines should be observed:

a. *Surfaces.* If possible, no rotational seating of mating parts should take place, since the gasket then has a tendency to become wrinkled or twisted, thus, destroying its sealing value. Sliding or rotating surfaces should be sealed in some other manner. Parallelism is required to achieve a uniform gasket squeeze and to minimize the area on which the pressure can act to extrude the gasket material from its seat.

b. *Joints.* The joint should be rigid. Gasket materials can take a permanent set (this is particularly true in the use of metallic gaskets) which limit their ability to seal if joint flexure is encountered. Thick flanges may be required that will not bow under pressure in the areas between bolt locations.

c. *Bolts.* If bolts are used to produce the desired gasket squeeze they should be strong enough to withstand the internal pressure forces which act to elastically elongate the bolts. Ref. 5 indicates that, after deducting the relieving effect of internal pressure, the residual gasket load should be 2 to 10 times the internal pressure. The higher value is used when personnel safety would be endangered by a gasket failure. Bolt spacing is equal in importance to bolt strength. Ideally, the gasket squeeze should be applied uniformly over the gasket surface. Since this is practically impossible in a bolted joint, as many bolts as are practical in the design should be uniformly distributed about the gasket periphery.

d. *Gasket Area-material.* The gasket area-material relationship should be such that the gasket can support the bolt preload without crushing.

e. *Finish.* The surface finish of the gasket seat should be as smooth as possible consistent with the gasket material and internal pressure. Generally, higher pressures require smoother surface finishes.

f. *Material.* The gasket material must be compatible with the working fluid and with the surrounding environment.

A large number of variables enter into the design of a gasketed joint; and, because these variables interact, a good seal presents a fairly complex problem. For a more detailed treatment of this subject, consult the references given at the beginning of this paragraph.

3-8.2. PACKING

A packing is defined in par. 3-7 as a dynamic seal used in situations where there is relative motion between adjoining rigid members, either axial or rotary motion. The design of the packing is such that an axial pressure on the packing is resolved into a radial pressure of the packing material against the moving surfaces to effect a seal against the transfer of fluids across the joint. A typical application of a packing seal is at the exit of the propeller shaft from the hull of an amphibious vehicle.

Packings are generally divided into two broad categories; compression packings and molded packings. Compression packings create a seal by being compressed axially between the throat of the stuffing box and the gland. The compressive forces on the packing force the packing material to flow radially outward to seal against the bore of the box and inward to seal against the moving shaft or rod. The packing material is lubricated to prevent friction, generated at the packing-rod interface, from scoring the rod. In the design of a compression packing the designer

should select a material that meets the following requirements:

a. Is sufficiently plastic to conform to the shaft and bore under gland pressures

b. Is compatible with the fluid being sealed and with the lubricants used so that no dissolving, swelling, or weakening of the packing material occurs

c. Is sufficiently plastic to conform to local static or dynamic rod deflections

d. Is not abrasive or corrosive to the rod

The compressive force acting on the packing must be adjustable in order to compensate for wear. The correct compressive pressure is an important consideration. Insufficient pressure will permit the seal to leak, and too much pressure will cause the packing to run hot, resulting in a shortened life. The correct packing pressure is determined by experience and experiment.

Packing lubrication is achieved either by using a lubricant soaked packing and replacing it periodically, or by a pressurized lubricating system. When a pressurized system is used, the lubricant should be introduced at pressures 5 to 10 psi above the operating pressure of the fluid being sealed.

Materials for compression packings are usually furnished with square or rectangular cross sections; although other confirgurations—such as nested wedges, cup and cone, and nested conical sections—are also available. These latter three types are illustrated in Fig. 3-9. Their action is such that axial compressive forces on the packing cause the elements to move radially with respect to each other to effect the seal.

A large and varied assortment of compression packing materials is available. It includes various fibers (vegetable, mineral, animal, and synthetic), metals (wire, foil, and other forms), and various lubricants, binders, and elastomers that impregnate the fibers to impart required properties. These are discussed in more detail in the references cited.

Molded packings as a group generally do not require gland adjustment after installation. Their design is such as to permit the pressure of the fluid being sealed to produce the force required to press the sealing elements against the wearing surface to effect the seal. This general classification of packings can be subdivided into two type categories, namely, lip- and squeeze-types. The lip-types include flange, cup, U-cup, U-ring, and V-ring packings. The squeeze-types include O-rings and other similar forms that rely on the interference built into the ring for effective sealing.

Molded packings are used mostly for sealing reciprocating motion. They are generally made of rubber, wax-impregnated leather, or rubber and fabric. Leather packings have low frictional characteristics, high tensile strength, and good extrusion resistance. They are limited, however, to temperatures below 200°F and to applications where they will not come in contact with acids, strong alkalies, or steam.

The following list gives some general design principles for molded packings:

a. A molded packing should be installed in such a way that it can expand and contract freely. The fluid being sealed should exert a sealing force against the packing during the "compression" cycle and should be relieved during the "return" cycle.

b. The packing should be adequately supported on the inside of the cup, but the support should provide clearance for swelling. A good rule of thumb is to provide a side clearance equal to one-third the packing thickness. All corners in contact with the packing, particularly on the inside support ring, should be well rounded to prevent chafing and cutting of the packing.

c. The metal sliding surface should be made of a hard, close-grained material with a fine finish to reduce packing wear.

d. Packings are seals and as such should not be subjected to bearing loads.

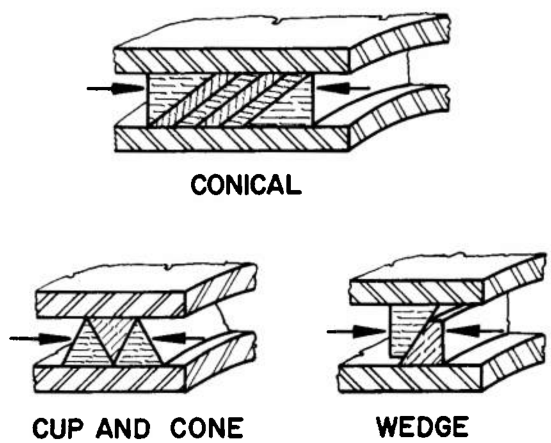


Figure 3-9. Compression-type Packing Designs

e. The packing should be protected from dirt, metal chips, and other contaminants to prolong packing life.

Squeeze-type packings, such as O-rings, delta rings, and T-rings, are widely used, economical, and efficient means of sealing static, rotating, and reciprocating joints. They work by controlled deformation, whereby the working fluid presses the packing into a confined annular space. Design considerations for squeeze-type packings require a balance between the initial packing squeeze (which adds system friction) and the maximum clearance permissible without extrusion. For O-ring packings, the recommended groove designs can be found in Refs. 32, 39, and 75. Squeeze-type packings should not be used for high-speed, dirty, or poorly lubricated applications.

3-8.3 LABYRINTH SEALS[5,32,40]

Labyrinth seals are designed to function between two moving parts by having a fixed clearance between the parts such that the restricted flow of fluid through this clearance creates the required pressure drop with a tolerable loss of fluid. The seal is constructed by cutting a series of annular grooves in the shaft, in the housing, or in both. The degree of sealing for a given liquid under a certain pressure is a function of the number of grooves and the clearance between the shaft and the housing. The sealing efficiency increases as the length of path of the leaking fluid increases and as the fluid flowing through the seal becomes more turbulent. Ref. 37 contains formulas that may be used in labyrinth seal design. The agreement between the formulas presented in the reference and actual leakage rates and pressure drops experienced, however, may not be very good. The achievement of a final working design usually requires some experimentation.

The advantages of labyrinth seals are reliability and simplicity. They are used mainly in applications where a relatively high leakage rate can be tolerated or where the pressure fluctuates rapidly or is pulsed, so that the labyrinth can offer a good dynamic restriction.

3-8.4 RADIAL SEALS[31,32]

A radial seal is commonly a rotating shaft seal in which the packing member contracts radially onto the shaft. This definition seems to apply equally to a compression packing. The main differences between compression packings and radial seals are the materials used and the design of the stuffing box. The sealing principles are the same for both types.

Radial seals are classed as felt radial seals and positive-contact radial seals. Felt seals provide a simple, reliable seal for shaft speeds up to 2,000 rpm. Felt has good resiliency to maintain a constant sealing pressure in spite of wear and end play, and it has a low friction coefficient ($\mu \approx 0.22$ when dry, 0.15 when presaturated with oil). Good design practice for felt radial seals includes:

a. The design should avoid excessive stretching of the seal when it is being installed.

b. The felt seal should not be fitted too tightly to the shaft nor unduly compressed by the retainer.

c. The height of the felt seal should be greater than its width to minimize distortion and permit firm clamping.

d. The felt seal should be a solid ring rather than split.

e. Felt seals should be easily removed and replaced.

A radial positive-contact seal is a device that applies a sealing pressure to a mating cylindrical surface to retain fluids and, in some cases, to exclude foreign matter. Radial positive-contact seals provide highly effective sealing over a wide speed-pressure-temperature range. They are categorized as cased seals, where the sealing element is retained in a precision-manufactured metal case, and bonded seals, in which the sealing element is permanently bonded to a flat washer or to a formed metal case. Seals of both types are provided with spring elements to increase the contact pressure against the shaft. Design considerations for positive-contact radial seals are:

a. Pressure range of conventional seals is in the order of 0-15 psi. Special designs are required for higher pressure applications.

b. Rubbing speeds up to about 2000 fpm are allowable for leather seals. Synthetic seals extend this limit to about 4000 fpm.

c. The temperature range is usually from -40° to 250°F for synthetic seals. Leather seals reduce the upper temperature limit to 200°F.

d. Experience indicates that shaft surface hardness should be at least Rockwell C30. If soft

metals are used for the shaft, it is advisable to press a hardened steel ring onto it to provide a running surface for the seal.

3-8.5 PNEUMATIC SEALS

A pneumatic seal is an inflatable seal. It consists of a tubular rubber strip molded with a "striker bead" on its top surface. The tube is molded in a collapsed position (Fig. 3-10) and is provided with an air inlet in its base. Low pressure inflation causes the collapsed tubular section to act as a diaphragm; and, since the cross-sectional area is quite small, the action is almost instantaneous. The striker bead moves outward against the sealing surface. Exhausting the pressure causes a rapid retraction of the striker bead to its low, collapsed position thus providing ample clearance for the sealing surface to be moved away.

This type of seal is available in various cross-sectional configurations, in strip and continuous loop form, to meet specific application requirements. Reinforcing fabrics such as cotton, nylon, Dacron, and glass fibers are incorporated to increase strength and durability. The polymers most commonly used for the manufacture of pneumatic seals are: natural rubber, neoprene, butyl, HYCAR*, silicone, and polyacrylate.

The pneumatic seal is particularly applicable in situations where adjoining parts meet without a precision fit yet must be capable of being sealed. Such situations are encountered when designing seals for cargo doors and personnel doors of amphibious cargo vehicles, the loading ramps of armored personnel carriers, and turret-to-hull seals of heavy armored vehicles required to have a deep-fording capability. An additional advantage of this type of sealing system is that it offers a means of sealing large door openings against water under pressure without requiring large door-closing forces.

In normal operations the seals remain in their deflated positions. Before the vehicle enters the water, however, the driver activates the inflation system, and an air compressor driven by the engine inflates the seals. Two drawbacks of this system are its relatively high cost, which must include the inflation system and its controls, and the need for the driver to remember to inflate the seals before entering the water.

*B. F. Goodrich Trademark

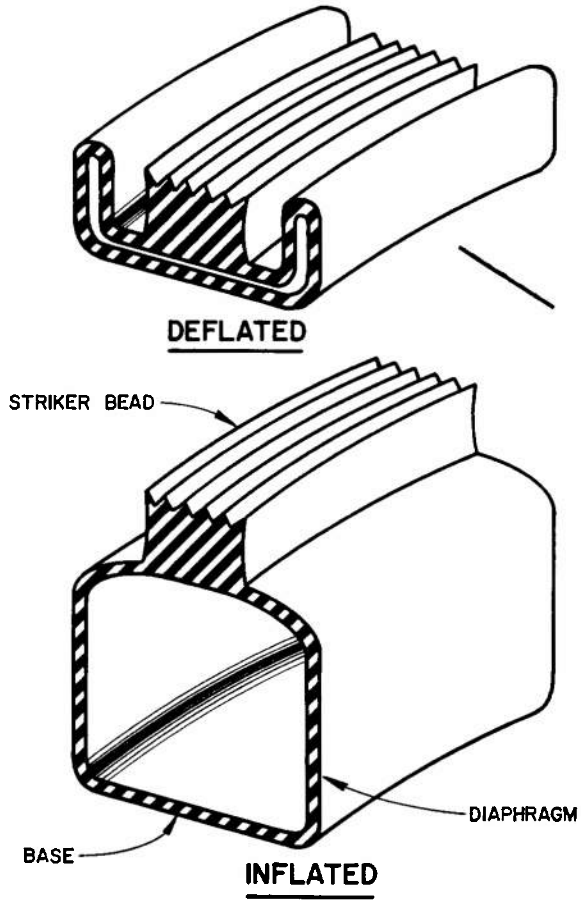


Figure 3-10. Pneumatic Seal

The following are some design considerations that affect pneumatic seals:

a. A pneumatic seal is primarily a static seal and, therefore, should not be used where there is a great deal of relative motion between the striker bead and the sealing surface during the inflated period.

b. The seal should be completely recessed into its retainer when in the deflated state to minimize chafing during normal cargo and personnel loading and unloading operations.

c. The seal should be totally confined by the door and the body or hull structure when the door is in the closed position. This will prevent the seal from "ballooning" into free spaces and allows the use of higher internal pressures when these become necessary.

d. When in the deflated state and with the door closed, the seal should not be overly collapsed to the total elimination of an air passage and should be free of obstructions,

kinks, wrinkles, and folds that would inhibit rapid inflation and might also result in permanent damage to the seal.

3-8.6 TURRET RING SEALS

Deepwater fording requirements impose a major sealing problem at the opening between the turret and the hull since this joint becomes subjected to hydraulic pressure. Amphibious vehicles have a simplified sealing problem because the turret ring seal is required merely to be water-resistant rather than watertight. A simple exclusion seal, such as an extruded rubber wiper, will prevent the entrance of water without adding a large rotational friction moment.

Turret ring seals on some of the older vehicles are similar to that shown in Fig. 3-11. This seal is not watertight. A seal adequate for deepwater fording is obtained by placing a triangular rubber strip in the turret-hull opening. The sealing strip is coated with grease, forced into the opening, and held in place with a strap or a band of elastic shock absorber cord. Newer vehicles, such as the M60 Tank, are provided with a turret ring seal such as that illustrated in Fig. 3-12. This seal consists of a wiper and a pneumatic seal. The wiper serves to exclude dirt and water from the turret bearing and the pneumatic seal is inflated to achieve a watertight joint. The pneumatic seal is discussed in the preceding paragraph.

3-8.7 MECHANICAL SEALS [31,32]

A mechanical seal is a shaft seal in which the packing member approaches the surface to be sealed in an axial direction. Mechanical seals are also known as axial seals or face seals. There are two basic types of mechanical seals—stationary and rotating. In a stationary seal, the sealing ring is in the joint housing and does not move. In a rotating seal, the sealing ring turns with the shaft. Mechanical seals are becoming more popular in the military environment because they have a very low leakage over their service life. Thus, the seal can be installed when the vehicle is manufactured and changed only when the vehicle is being overhauled. Other sealing advantages are:

a. Reduced friction and power losses

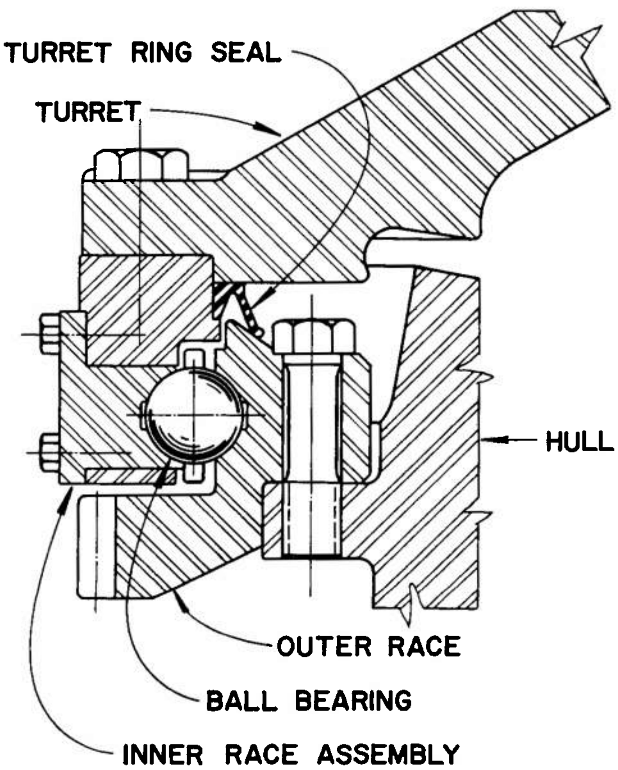


Figure 3-11. Turret Race Assembly, M48 Tank

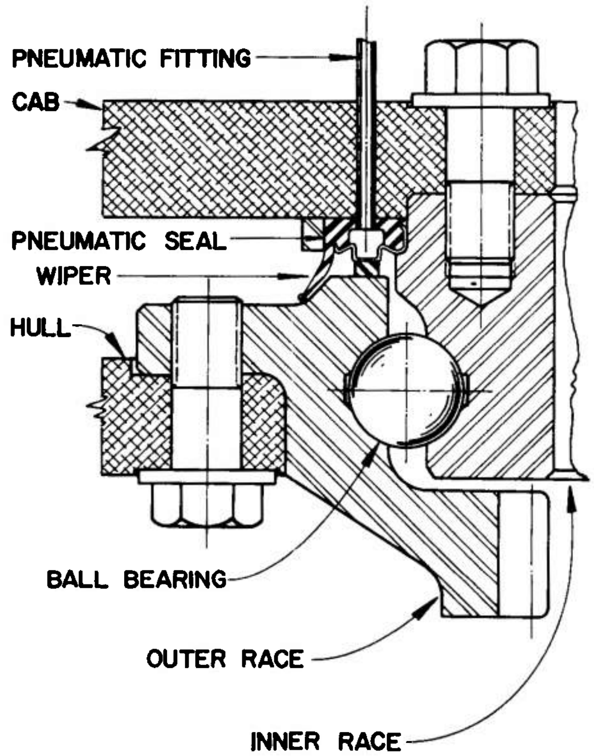


Figure 3-12. Pneumatic Turret Ring Seal on Self-propelled Howitzer

b. Elimination of wear on shaft or shaft sleeve

c. Relative insensitivity to shaft deflection or endplay.

d. Freedom from periodic maintenance.

Mechanical seals do, however, require precision manufacturing and careful installation. Mechanical seals have been used for sealing pressures up to 3,000 psi, for speeds up to 50,000 fpm, and for temperatures from −425°to +1,200°F. Special seals can be designed to exceed these ranges. The primary application for mechanical seals in military vehicles is in the suspension system rather than in the design of bodies and hulls. Therefore, no more will be said on this subject here. For further discussion consult the references cited.

3-8.8 OTHER SEALS [32]

Most prominent in the "other seals" category are diaphragms and exclusion devices. Diaphragm seals form a continuous membrane across the fluid path and are used in applications where the relative motion between the two members of the joint is reasonably small. In military vehicles, diaphragm seals are used at the gun-turret juncture of tanks and self-propelled guns and at vehicle bulkheads to seal control rods and cables against the entrance of dust, dirt, and moisture. In future vehicles which may be required to withstand CBR attack, diaphragm seals may be required to prevent the leakage of air. Some factors that should be considered when designing diaphragm seals are:

a. The relative movement between the two members of the joint. The diaphragm must not be stretched unduly at the maximum motion positions, nor should it be kinked or chafed at any normal operational position of the joint.

b. The diaphragm should be adequately sealed to each joint member in a manner that permits easy removal of the diaphragm for vehicle maintenance or servicing.

c. The diaphragm material must be capable of withstanding the military environment, and it must be impermeable if pressure sealing is required. It must also be sufficiently flexible to cause no undue resistance to joint motion.

Exclusion devices—which include wipers, scrapers, and boots—resist the entrance of solid or liquid foreign matter. Normally, an exclusion device is used where the foreign material is not under pressure and an absolute seal is not necessary. Wipers are commonly used on reciprocating surfaces, such as roll-down windows, and resemble lip seals. The lip makes contact with the surface and wipes off excess dirt or moisture. Scrapers are essentially wipers with a metallic lip that scrapes heavy or tenacious material, such as mud, from reciprocating surfaces. Boots are removable covers that are fitted over exposed components such as gun barrels, winches, or cylinder rods for protection from the environment.

3-9 APPLICATIONS

3-9.1 DOORS, GATES, AND ACCESS OPENINGS

Seals for doors, gates, and access openings are generally made from extruded or molded rubber; and their shape and size depend upon their intended use. If a door seal is to be moisture-resistant only, it may be made in a simple shape and require only a small amount of squeeze to achieve adequate sealing. On the other hand, if the seal is required to prevent the passage of very small particles or water under pressure, it may be necessary to place a seal on both the door and the door frame. The seal shape may have to be complex to form a tortuous path, and the door design may also be complicated if the squeeze forces required to achieve a seal are large enough to require a complex closing and locking mechanism.

One method used to reduce the door squeeze force at closing is by means of pneumatic seals. These seals are described in par. 3-8.5. The door, gate, or access cover is closed while the seal is unpressurized. A strong seal squeeze is then achieved by inflating the seal. While this sytem is complex compared to a normal static door seal, the pneumatic seal is effective and allows a low door closing force. The closing force becomes appreciable for large doors equipped with conventional seals, so that pneumatic seals have their greatest applications on vehicles with large personnel and cargo doors and with gates that require watertight seals.

Door seals are installed by bonding the seal directly to the door or door frame or by mechanical means. When mechanical attachment is used, bearing plates should be provided at the points of attachment to distribute the attachment load over a significant seal area. Care should also be exercised to insure that the means

of seal attachment does not interfere with the sealing capabilities of the seal or with its useful life.

3-9.2 WALL AND FLOOR PANELS [38,40,41]

Sealing compounds are used to seal wall and floor panels and to isolate them from vibrations and shocks. Where both sealing and vibration isolation are required, the sealing compound is placed between the mating panels. If only sealing is required, the sealing compound is placed on the inside or outside surfaces of the joint, depending upon whether leakage to the exterior or interior of the vehicle is to be prevented. Ref. 38 lists many of the compounds that are available for sealing and discusses their methods of application.

In selecting a sealant, it is important to consider the elastic lifetime properties of the material, its ability to adhere to the type of surface being used, its ability to withstand the operating environment, its ease of application, and its ability to seal. The joint design must also be considered in applications in which the material being sealed is under pressure, such as in vehicle joints that are submerged in water. The sealing material should always be placed on the wetted side of the joint, and the wetted area of the sealing compound should be greater than the open area of the joint so that any differential pressure acting on the sealant will tend to force the sealing compound into the joint opening. To facilitate this action, the designer may provide a shoulder at the joint to act as a bearing surface for the sealant. On lap and angle joints, the joint edges should be beveled inward, if possible, to make the seal more effective.

3-9.3 WINDOWS

Fixed windows, i.e., windows that do not open, are sealed by using an extruded or molded rubber strip between the glass and the window frame and applying an adhesive sealant on the outside to seal the juncture of window-seal and seal-frame to insure watertightness. The molded rubber sealing strip is attached to the vehicle by either a bonding compound or by mechanical means. The bonding compound is preferred because it also acts as a sealant between the molded strip and the vehicle.

Windows that swing open, such as those used in some vans, are permanently sealed into their frames by the method just described. The frame is then sealed with respect to the body or hull in the same manner as is a door. Windows that slide up and down are sealed to a frame that slides in a trough formed in a molded or an extruded rubber strip. The sliding seal achieved in this manner is weather-resistant rather than weatherproof.

The window seals on windows mounted in doors must serve as vibration and shock isolators in addition to their function as weather seals. These windows should be supported as uniformly as possible around their peripheries to reduce stress concentrations in the glass. Window frames for roll-up windows in doors should be adequately supported to prevent glass breakage when the door is slammed, particularly when the window is in a partially open position.

3-9.4 CABLES, HOSES, AND CONTROL ELEMENTS

Hoses and ducts are sealed to prevent undersirable leakage of gaseous or liquid contents. Sealants compatible with the contents are employed at all joints, and clamps and supports are used to insure a tight seal. The clamps should provide a uniform squeeze around the periphery of the hose or duct. Aircraft-type hose clamps are generally used to secure hoses because they offer a means of attaining an adjustable, uniform squeeze.

Seals are placed around cables and control elements where they pass through the bulkheads from the crew compartment into the engine compartment or from the inside of the vehicle to some component on the outside. The method of sealing depends upon the level of sealing required and the freedom of movement required for control rod or cable action at the bulkhead, floor, or wall. When only a heat seal is required, a simple, loose fitting, felt or rubber grommet is used that serves as both an adequate heat seal and as a rubbing pad. When an air or moisture seal is required and the amount of cable or control rod travel is small, a relatively simple bellows seal may be used effectively.

The sealing problems associated with long stroke cables and control elements increase as

the degree of isolation desired increases. Seals against small particles, air, gases, or water pressure require a close fit and an appreciable seal squeeze. These sealing requirements are in direct opposition to ease and physical freedom of movement. Some systems may require the use of power boosters to overcome the resistance developed by the seals.

3-9.5 ACCESSORIES

Accessories—such as lights, instruments, vision devices, antennas, etc.—are normally prepackaged so that the body or hull designer needs to consider their sealing requirements only as they affect each particular installation. Some accessories, such as radio masts, require openings in bulkheads or in the exterior structure of the vehicle. These are sealed with adhesive sealing compounds or gaskets. In these installations, the seal should also act as a vibration and shock isolator to reduce the noise that might otherwise be transmitted into the body or hull by the accessory.

Some lights contain removable light bulbs and are sealed to prevent the reflector from collecting dirt or becoming corroded by moisture. These seals are rubber or fiber gaskets which may be bonded into place by an appropriate adhesive[42].

SECTION IV—CABLE ROUTING

3-10 ELECTRICAL CABLES [76]

The electrical cables used in military vehicles are coated to protect them from environmental effects such as dirt and moisture. It is the responsibility of the designer to so route and secure the electrical cables in the vehicle that their protective coatings are not subjected to a harsher environment than that for which they were designed. In particular:

a. Cables should be bundled. It is much easier to adequately route and support a bundle of cables than to try to route each electrical line separately. Particular electrical leads can be branched from the bundle in the vicinity of the component at which the lead is required.

b. Cable bundles and leads should be routed in a manner than will provide the maximum protection against moisture, dirt, snagging, and abuse. Closed or partially closed structural sections provide excellent cable routes. When the cable route is exposed to water, adequate drain holes should be provided to prevent water accumulation and water damage.

c. Electrical cables and leads should be protected from vehicular vibrations and travel shocks. Supporting clips and brackets should be coated with rubber or provided with rubber grommets to eliminate chafing. They should contain the cable or lead with a firm hold and be so spaced that the cables are not taut and yet do not sag excessively.

d. Electrical cables should be so routed that they will not be exposed to either excessive heat or cold. In particular, the cables should not be run close to the exhaust system because the excess heat will cause premature aging of the electrical insulation.

e. Terminal blocks should be used at bulkheads to facilitate servicing and sealing. By using terminal blocks, components can be easily removed, serviced, and replaced with minimum disturbance to other equipment. If a compartment is to be sealed against air leaks, the terminal offers a means of achieving a permanent seal without disturbing the electrical serviceability of the vehicle. However, when terminal blocks are used, the electrical contacts must be firmly made and retained. A mechanical connection that seals against moisture and dirt should be used for this purpose.

f. Access openings should be provided along the cable route. Servicing and repair of damaged cables should be possible without having to remove and replace large sections of the electrical cable system. Each branching lead should be readily accessible so that, in emergencies, electrical components can be quickly replaced.

3-11 BRAKE LINES

When planning the routing of brake lines, the body or hull designer should consider the

following factors:

a. A protected route that prevents the snagging or impact of brake lines on large objects. The military environment is such that the vehicle is expected to spend a significant period of its operating life under cross-country conditions. Under these conditions, the vehicle's bottom can be expected to be dragged through mud and over large objects, such as rocks and logs. To minimize damage to the brake lines, they should be routed inside the vehicle structure to the maximum extent possible. Where outside routing is required, the vehicle structure should be used to shield the lines by running the lines inside of channels or closed body, hull, or frame sections. Where appropriate, guards should be installed which are strong enough to shield the lines from impact.

b. Support mountings for the brake lines. Use a type that reduces and distributes the travel vibration and shock loads experienced by the lines. Rubber-coated clips and grommeted support brackets should be provided to support the brake lines along their routes. The supports should be so spaced that the lines will not come in contact with the adjacent vehicle structure when they are deflected by travel loads. Supports should also be provided at fittings so that brake line deflections will not impose undue loads on the fittings.

c. Freedom of action is required at the brake drum. Because the brake drum has a relatively large range of vertical motion due to the action of the suspension system, the brake line should be free to flex to accommodate this motion. While freedom of action is required, it should not be accompanied by an increase in brake line vulnerability to road damage. The flexible coupling should be positioned in a protected portion of the suspension system, and the suspension system structure should be used to provide a shield for the brake line as it is run to the brake drum.

d. Access openings should be provided along the route of the brake line where it would otherwise be inaccessible. Access is required to permit repair or replacement of brake line sections and support clips that may become damaged by aging, accident, or battle.

3-12 CONTROL LINES

Control line routes should have the following attributes:

a. A direct route. Each change in direction that a control lever or cable system must take adds to the system complexity and operating power requirements. If a complex route is required for a mechanical control line, an electrical control system should be considered as an alternate.

b. Freedom of motion. The control line route must be free from static and dynamic interferences both externally and internally. Control lines must be run along paths that are free from other equipment, such as cables that can inhibit the control action by deflectional interference under road shock loads. Long control rods and thin bell cranks also can deflect under dynamic loading and cause binding. If long control rods are required they should be supported by free running fittings which can limit any dynamic deflection that may be induced. Thin bell cranks should be stiffened by beads or formed sections, or the bell crank will be limited or prevented from causing binding.

c. Protection from moisture and dirt. Where practical, control line routes should be located where exposure to moisture and dirt is minimized. If exposure is unavoidable, bushings and control-cable ends should be sealed or easily inspected and cleaned.

d. Access. Control rods, bell cranks, cables, pulleys, and fittings should be readily accessible for servicing and repair. Control lines should be capable of being serviced, repaired, or replaced without the need to disassemble or remove any vehicle component other than the access doors.

SECTION V—VISION DEVICES

3-13 LIGHTS AND LIGHTING

The lighting system found on most modern military vehicles consists of the following:

a. Two headlights for illuminating the road ahead of the vehicle.

b. Two parking lights for indicating the location of the vehicle when parked.

c. Taillights to illuminate the rear license number plate and to show red lights to the rear.

d. Body lights, such as dome and step lights, and other lights necessary to illuminate the interior of the vehicle.

e. Instrument panel lights to illuminate the instruments.

f. Special lights, such as backing lights to illuminate the road to the rear when backing, signal lights to indicate change of direction, stoplights to indicate the application of brakes, clearance lights to indicate a large vehicle's width or length, and spotlights, floodlights, and searchlights when needed on special vehicles.

g. Blackout lights to enable vehicles to move at night without being observed permit normal performance of all operational and combat tasks by the crew. This includes blackout taillights, a blackout stoplight, and a blackout switch all interconnected with the regular lighting system in a manner that prevents the accidental energizing of the regular lights when the blackout lights are on. A detailed discussion of this circuitry is given in Ref. 43.

All exterior lights (except blackout lights) should confrom to ICC Regulations[44]. They should be either hermetically sealed or waterproof and should be provided with protection against physical breakage, shock, and vibration by design, location, guards, and elastomeric housings and mountings. Headlights should be guarded and recessed. Identification, side-marker, and clearance lights should be installed at a height of not more than 9 feet from the ground. Turn signals, stoplights, and taillights should be SAE-S-AI-T, 4-inch nominal diameter type. Lights and reflectors should not be mounted on rub rails, vehicle bumpers, hatches, or hoods. Military Standard (MS) lighting components and assemblies, which have been proven serviceable, are available[45].

Turn signals should be an ICC-SAE turn signal system (Refs. 44 and 46) and should be readily accessible to the driver. (The turn signal actuator is normally mounted on the steering column.) Visual flasher operation should be indicated by green left- and right-turn indicator lights mounted on the instrument panel. In addition, an emergency flare control—for flashing all turn signal lights during emergencies—should be provided which should include a Class A flasher separate from the turn-signal control flasher.

Special-purpose exterior lights, such as floodlights or spotlights, should be mounted where required for best illumination. These and their mountings should be waterproof and protected by suitable guards from impact damage.

Interior lighting includes instrument panel illumination, various warning and indicating lights, and general crew compartment and fighting compartment illumination. For optimum retention of eye adaptation to darkness (night vision), only red lights should be visible in the fighting compartment under nighttime combat conditions[47]. The intensity of instrument panel lights should be controllable by means of a suitable dimming switch. Dome light illumination should have sufficient intensity to permit normal performance of all operational and combat tasks by the crew. This includes such tasks as service of the weapons; use of all controls without searching, groping, or reliance on habit patterns; reading of all instruments, dials, and gages normal to operating the vehicle; and the reading of maps, instructions, orders, etc. Excessive illumination should be avoided in order to prevent detection by unfriendly forces. White dome lights should be also provided for use in noncombat situations, but with suitable safeguards against their unintentional use. Warning, "on-off", and other circuits should be clearly visible but not overly bright. Because a red light jewel designed to be visible in daylight may be overly bright and glaring at night, provisions should be made for varying the intensities of such lights. The general level of light intensity on all on-board facilities and equipment required for proper functioning of the crew should be reasonably well-balanced to avoid eye accommodation or adjustment when the view is shifted from one component to another.

Van-type vehicles (including trailers)—such as mobile command posts, communication centers, ambulances, medical vans, and similar vehicles designed to provide working or living quarters for personnel—require white interior lighting. Some amublances are equipped with surgical lights, and the interiors of some large vans are illuminated with fluorescent lights. All vehicle lighting is designed to operate from the standard military 24-volt DC system. Most van-type vehicles are also equipped to operate on 115-volt AC service from an exterior plug-in source.

3-14 WINDOWS AND PORTS

An analysis of the field of view required by each crew member is needed for effective design of windows, ports, vision blocks, and periscopes (also see pars. 3-15 and 3-16). When designing windows and ports in tactical vehicles, chief concern is for the driver's vision and the extent to which his visibility is affected. Drivers of these vehicles should have maximum possible vision for safe operation. As a minimum, the driver should have generally unobstructed view from straight ahead to 90° to either side, excluding the effect of passengers. Interference by windshield posts or supports should be nominal and capable of being overcome by a simple movement of the head[48].

All glass windows should be safety glass in accordance with ASA-Z26.1a—1964[49]. The windshield should be of laminated safety glass (Type ASI) and should be of the knockout type with elastomeric mounting. All other windows should be AS2-type glass. Window construction should conform to ICC Regulations[44].

Ports are slits or openings in the sides, front, and back of vehicles through which weapons can be fired by the vehicle crew or by its passengers. Two vehicles currently under development that are equipped with ports are the XM701 Mechanized Infantry Combat Vehicle (MICV), and the XM706 Armored Car, also known as the Commando Escort Vehicle. These are shown in Figs. 3-13 and 3-14. The concept of the MICV will enable the infantry rifle squad to fight on the move, firing their small arms from the ports in addition to a cupola-mounted 20 mm gun and 7.62 mm machine guns. The XM706 Armored Car is a wheeled vehicle with similar capabilities. The capability of firing through ports gives the squad an effective means of repelling close in swarming-type attacks.

The number and size of ports should be minimized, as each constitutes an opening in the hull that is vulnerable to bullet splash and the entrance of projectiles and fragments. At the same time, however, they must be adequate in number, size, and location to permit overlapping and mutually supporting fields of fire. The port opening is usually made just large enough to accommodate the weapon for which it is

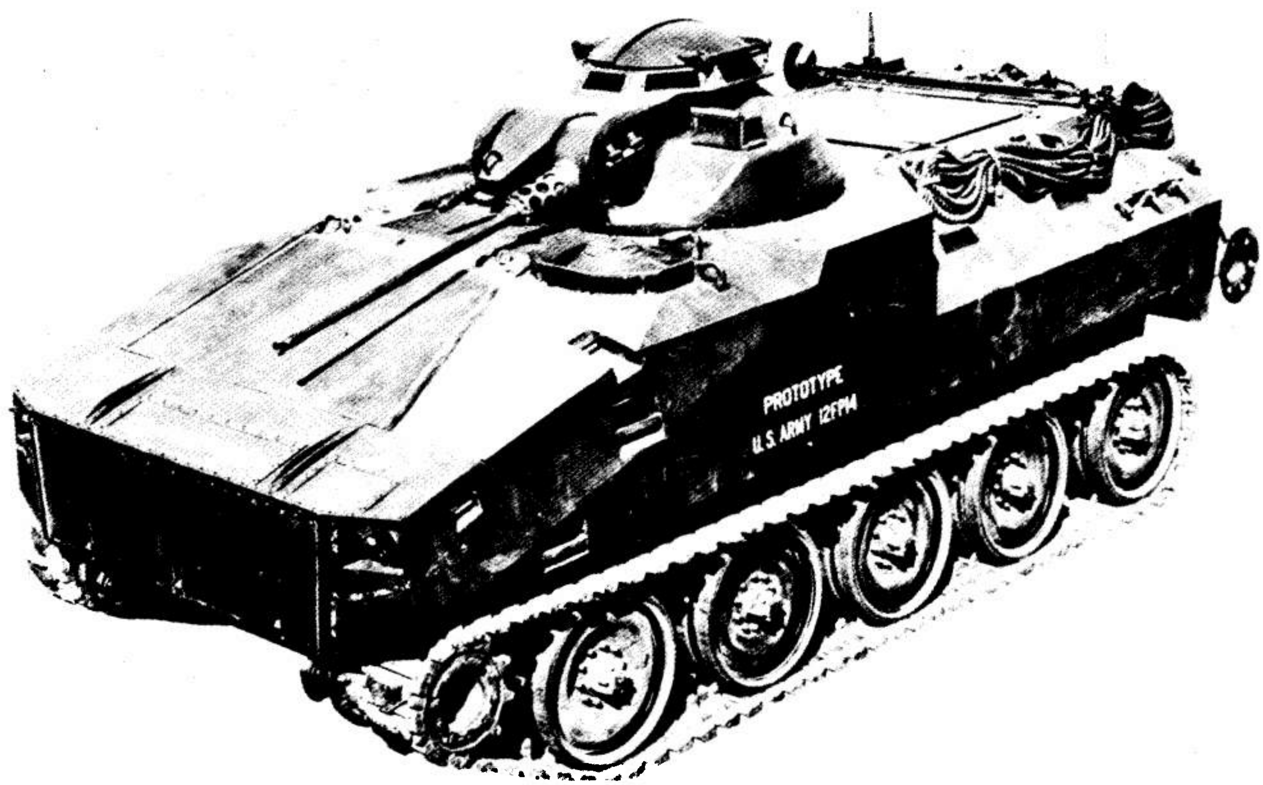


Figure 3-13. Mechanized Infantry Combat Vehicle, XM701

Figure 3-14. Armored Car, XM706 (Through courtesy of Cadillac Gage Co.)

designed. Aiming of the weapon and observation of fire is usually done through separate vision blocks or periscopes.

Firing port closures or covers should be made of the same material as the hull in which they are mounted, or at any rate, they should afford the same level of protection when closed as does the adjacent hull wall. Closures may be designed to rotate laterally about a pivot, slide, or be hinged. Rotating and sliding closures require the least amount of space but are more vulnerable to being immobilized by burring, keying, and deformation (see par. 2-12.3.5). Hinged closures are generally more satisfactory when made to swing outward rather than inward, and hinges should be an integral part of the cover to eliminate the possibility of being pried away by a projectile. Rubber bushings and washers are often incorporated into the hinge to compensate for deformation of hinge pins and burring of bearing surfaces.

All closures should be provided with bullet deflectors and protection from bullet splash (see par. 2-13.3.4 and Fig. 3-5). The inside supporting structure for the closure must be designed to withstand the shocks of projectile impacts and blasts that may be experienced by the closure. All closures should be capable of being locked from the inside of the vehicle.

Heavily armored vehicles are often equipped with vision ports that can be opened when the vehicle is not engaged directly in combat. When opened, these offer the best visibility; for vision through them is direct. These ports are closed, when under fire, by heavy armored covers that are hinged on the outside and secured on the inside by a locking bar. All viewing is then done through indirect viewing devices, such as vision blocks and periscopes.

3-15 VISION BLOCKS

Vision blocks are used in armored vehicles to give the crew a means of observing outside the "buttoned up" vehicle with some degree of

protection against projectiles, projectile fragments, and other missiles. They are laminated of bullet-resistant, shatterproof glass, assembled into suitable casings, and mounted in the turret cupola. Their principal disadvantage is that a few impacts against the glass by small caliber projectiles generally causes sufficient cracking to render the block worthless as a viewing device. Therefore, several vision blocks should be installed with overlapping fields of view to provide some redundancy to the system. Their vulnerability can be reduced by making them narrow vertically so that they resemble viewing slits, and by setting them at ballistically advantageous angles to minimize the probability of impacts normal to the glass. In addition, vision blocks should be located where they will not be exposed to projectiles or bullet splash ricocheting from other surfaces of the vehicle. On certain World War II German tanks, armored hoods very effectively protected vision blocks from fragments, blast, and even small arms attack.

A common practice in tank design is to equip the commander's cupola with sufficient vision blocks to provide the commander with 360° vision and a $-15°$ to $+65°$ vertical field of view.

Violent impacts on vehicle walls in which vision blocks are mounted can cause these devices to crack. Protection against such shocks can be obtained by providing a rubber liner around the casing of the vision block to serve as a shock isolator.

3-16 PERISCOPES

Periscopes provide the best means thus far available for safe vision while under fire. Their principal advantage is that they can be mounted where they are least likely to be damaged by attack and, if damaged, can be easily replaced. The mountings, fastenings, and working parts of periscopes are generally highly susceptible to damage and immobilization by high shock; therefore, unless necessitated by other design considerations, the periscope should not be mounted to a wall that is exposed to direct attack. The best overall combination of protection and visibility is afforded by mounting the periscope to the roof of the turret or hull. Here it can be made adjustable in both azimuth and elevation if necessary, the moving parts are hidden from direct attack by ground fire, and the location is amenable to withdrawing and replacing the periscope from within the vehicle when necessary—even when under fire. The portion of the periscope that extends above the roof is usually made of brittle materials to assure their shattering and breaking away under impact rather than deforming and becoming inoperative. The installation design should, therefore, provide for easy replacement from inside the vehicle.

Like most vision devices, periscopes require that a direct opening or hole be provided through the armor and thus compromise the armor design. As a means of offsetting this weakness, armored hoods are sometimes placed over the exposed portions of periscopes to provide increased protection against blast, fragments, and small arms fire; however, they do have the disadvantage of limiting vision.

Figs. 3-15 through 3-19 show several types of periscopes used in armored vehicles. Those shown in Figs. 3-15 and 3-16 are made of solid plastic and have a single power of magnification. The former (Fig. 3-15) is mounted around the turret cupola of the M48 Medium Tank for use by the tank commander, and the latter (Fig. 3-16) is mounted around the driver's hatch

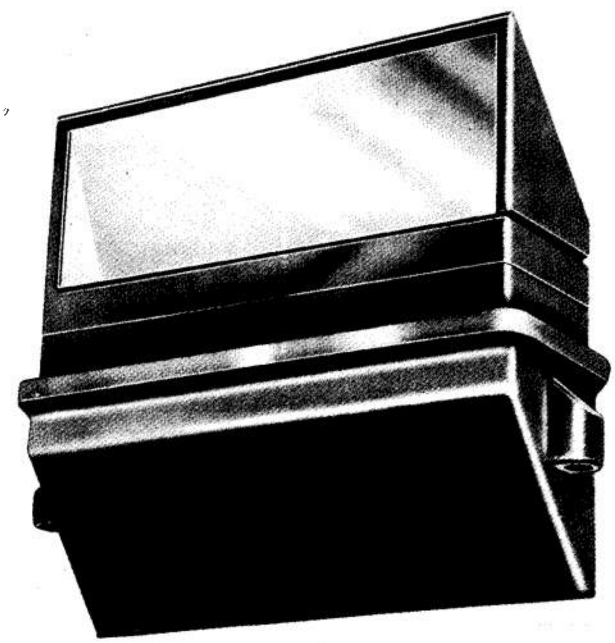


Figure 3-15. Single-power Periscope, M17

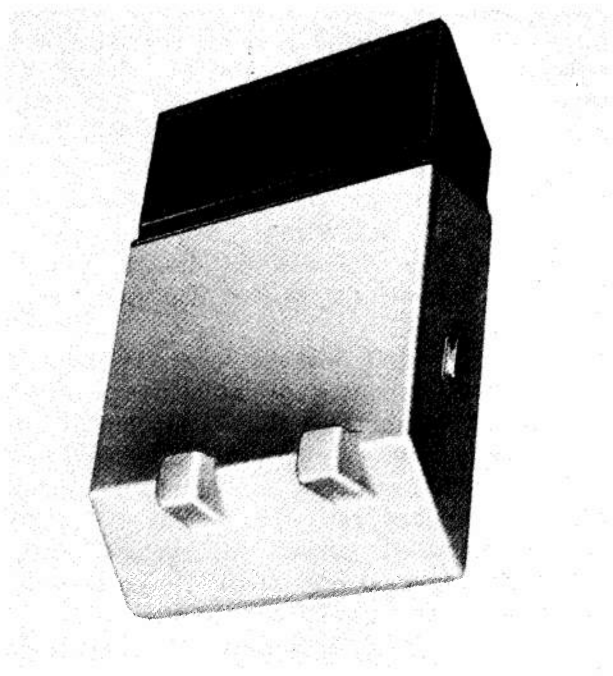


Figure 3-16. Single-power Periscope, M26

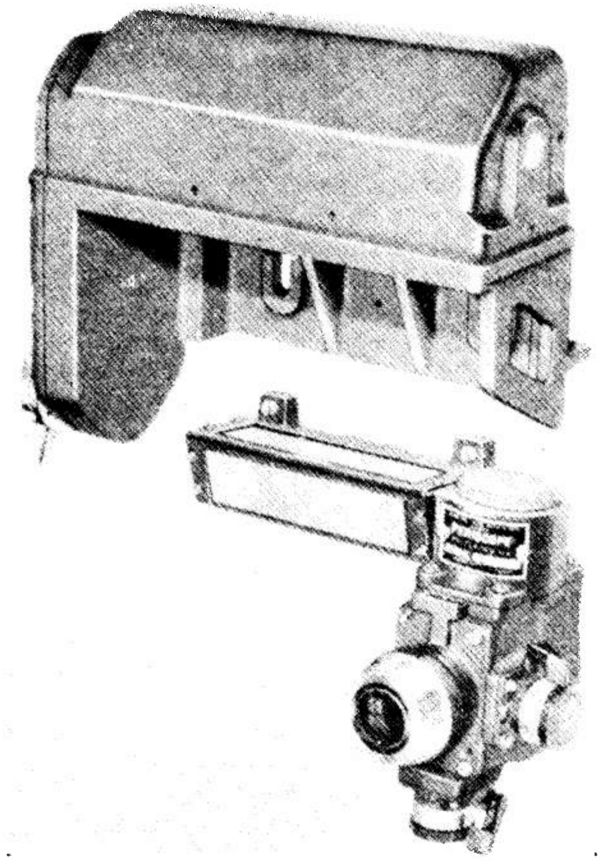


Figure 3-17. Dual-power Periscope, M20 Series

opening for use by the driver. A minimum of three of each are generally used to give an adequate field coverage.

Fig. 3-17 shows the M20 Periscope, and Fig. 3-18 shows an installed view of the same device. This is a monocular instrument with two built-in optical systems—a single-power system for wide angle close observation of terrain and a six-power system for sighting distant objects and for ranging on targets. It consists of two separate major components—the head assembly and the body assembly, as shown in Fig. 3-17.

Fig. 3-19 shows an infrared viewing periscope for use in night driving. It is a single-power binocular instrument designed to convert invisible infrared rays to a visible image which is viewed through conventional eyepieces. Other periscopes of this type are available that have dual optical systems—a single-power and a seven-power.

Details of the various periscopes that are available can be obtained from Technical Manuals covering each particular instrument.

3-17 MIRRORS

Tactical tracked and wheeled vehicles are equipped with outside rear view mirrors. The outside mirror construction is governed by the regulations set forth in Ref. 50. The size and location of the mirrors are determined by the vehicle designer or by the vehicle specifications. Normally, one mirror is mounted on each side of the cab in such a manner that an adequate rearward view is available along both sides of the vehicle. When determining the adequacy of mirror view, the rear window should be blocked so that adequacy of view for passing and backing (particularly with a trailer) can be determined based solely on the capabilities of the mirror system.

3-18 BLACKOUT CONSIDERATIONS

3-18.1 LIGHTING REQUIREMENTS

Military vehicles are required to have a set of blackout lights which includes:

a. Blackout drive lights. These lights are normally mounted near the headlight clusters and are used to illuminate the ground immediately in front of the vehicle.

b. Blackout tail and stop lights mounted on the rear of the vehicle.

c. Blackout marker lights mounted as needed to outline the vehicle.

d. Blackout instrument lights. These lights are red to retain the adaptation of the eyes to darkness.

In addition, the vehicle may be equipped with infrared driving lights, mounted in the headlight clusters, which have high and low beam filaments in each unit. These headlights project an infrared beam and must be used in conjunction with an infrared receiver. Ref. 51

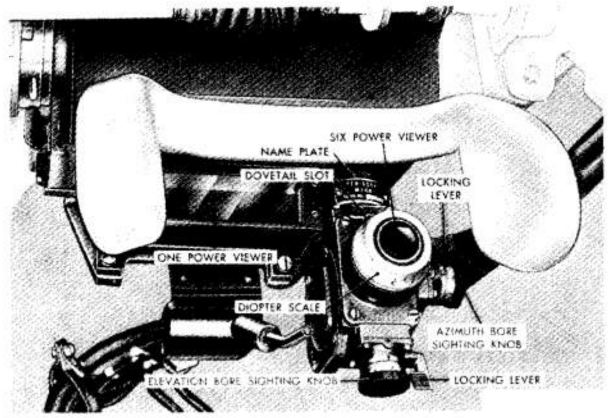


Figure 3-18. Installed View of M20 Series Periscope

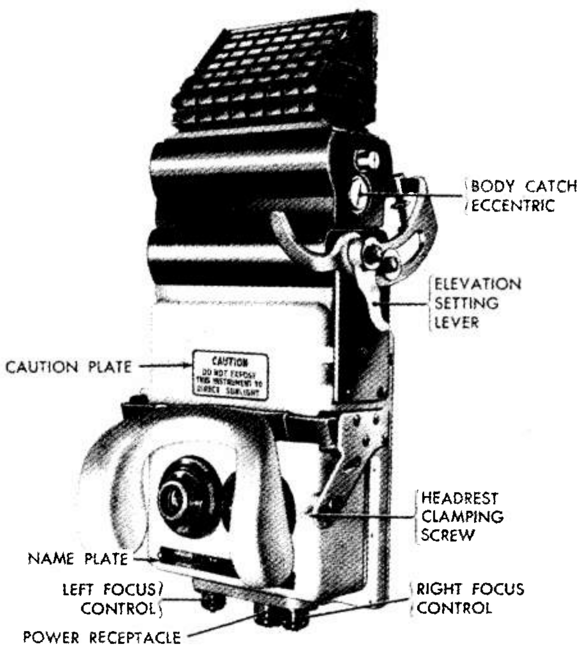


Figure 3-19. Infrared Viewing Periscope, M24

indicates that a substantial gain in blackout driving can be obtained by using infrared headlights as compared with using normal blackout driving lights under cross-country driving conditions. The speed capability, using infrared lights, seems to compare favorably with that attainable by using regular headlights.

It is important that the blackout light switch be so constructed that all of the regular night service lights are locked out of the circuit when the blackout lighting system is energized. This lockout is required to make accidental illumination of any other light impossible. The standard MS-51113 vehicular light switch may be used for this purpose. The van or closed command type vehicle is an exception to this rule. In these vehicles, the interior working compartment is on a separate lighting circuit that has its own blackout light switch which locks out the other interior lights when the interior blackout light is activated.

3-18.2 SEALING REQUIREMENTS

Concealment has always been a major contributing factor to the element of surprise on the battlefield as well as a major element in passive defense. For nighttime operations, exterior lights can be extinguished and replaced with special blackout lights and infrared viewing devices; but interior lighting is necessary in many types of vehicles, particularly in many van-type vehicles, command and communications vehicles, and medical vehicles, as well as in combat vehicles. Leakage of this light to the exterior at night calls attention to the vehicle and compromises any tactical advantage that it might otherwise have. For these reasons, every effort should be made during the vehicle design phase to prevent light leakage.

The obvious sources of light leakage are from around doors, windows, and hatch covers. Less obvious sources, but equally important, are joints between wall, roof, and floor panels; ungasketed inspection and service plates; openings around control rods, electrical cables, and hydraulic and pneumatic lines; ventilator intake and exhaust ports; joints between turrets and hulls; joints around the closures of weapon ports, particularly around the mantle openings provided for the primary and secondary armament; and joints around viewing and

ranging devices (periscopes, viewing blocks, telescopes, range finders, and aiming sights).

Interior light will be transmitted outward through optical viewing devices. Therefore, combat and tactical vehicles equipped with these devices should be provided with a combat interior lighting system (orange-red) in addition to the conventional white light. In no case should lights be so located within the vehicle that they will shine directly into a viewing device.

Fortunately, seals that are normally used to protect military vehicles from the entrance of dust, weather, water, and CBR contaminants are usually adequate for sealing against the leakage of light. The general subject of seals is discussed in pars. 3-7 through 3-9. However, the doors of command- and communication-type vehicles and of certain vans that require interior lights during operations are necessarily used with considerable frequency even in night operations. In order to prevent interior light from spilling over the landscape every time a door is opened, the entryways of these vehicles must be provided with blackout curtains, screens, or baffles so arranged with right-angled or return passages as to prevent the escape of light when the door is opened. These light seals may be temporary structures, attached or detached as needed; or they may be a permanent part of the vehicle. The interior of the passageway should be rough textured and have a dark, nonreflective color to minimize the reflecting of light around the baffles. Top, bottom, and sides must be made light tight. Canvas is the material most commonly used for entrance light seals; although rigid structures are also used. When designing light tight entrance tunnels, due consideration should be given to the dimensions and space requirements of all equipment that will be moved through it. For example, a blackout entrance tunnel to a hospital van would need to be sufficiently large to accommodate a litter and bearers.

SECTION VI—ELECTRICAL DEVICES

Electricity and electrically operated devices play an important role in the operation of modern military vehicles. Although these are not the direct responsibility of the body and hull designer, their numbers, size, and operating requirements make it imperative that they be considered throughout the design phase. The inspection, service, repair, and routine maintenance of electrical devices, or any device, can be accomplished much more efficiently when they are integrated into the system as part of the original design by designers who were fully appreciative of the requirements of these items than when they were installed after the body or hull design was nearing completion.

It is not the intent of this section to give a presentation of the fundamentals of automotive electrical systems nor to discuss the fundamentals, design, and operating principles of the many electrical devices found in military vehicles. A good introduction to this subject is given in Refs. 52-56, and detailed coverage of any specific vehicle or device can be obtained by consulting the appropriate Technical Manuals[57]. The objective of this section is to acquaint the designer with some of the electrical devices that are more or less unique to military vehicles and to indicate their influence on body or hull design.

The electrical systems of wheeled and tracked vehicles are generally similar and consist of a number of interconnected systems. The hull wiring of a combat tank, for example, comprises the battery and generating system, starting system, horn and lighting system, engine ignition system (consisting of switches and cables to engine wiring junction box only), auxiliary generator and engine electrical system, personnel heating electrical system, warning lights system, and hull radio and interphone system junction boxes and interconnecting cables. These are generally interconnected as shown in Fig. 3-20. The electrical system of a combat tank turret comprises the turret traversing and gun elevating control system, fire control and safety system, interference switch system (automatically elevates gun to clear fenders, fender-mounted equipment, and rear deck during power traverse to the rear), radio feed and interphone system, and turret accessories systems. A wiring diagram

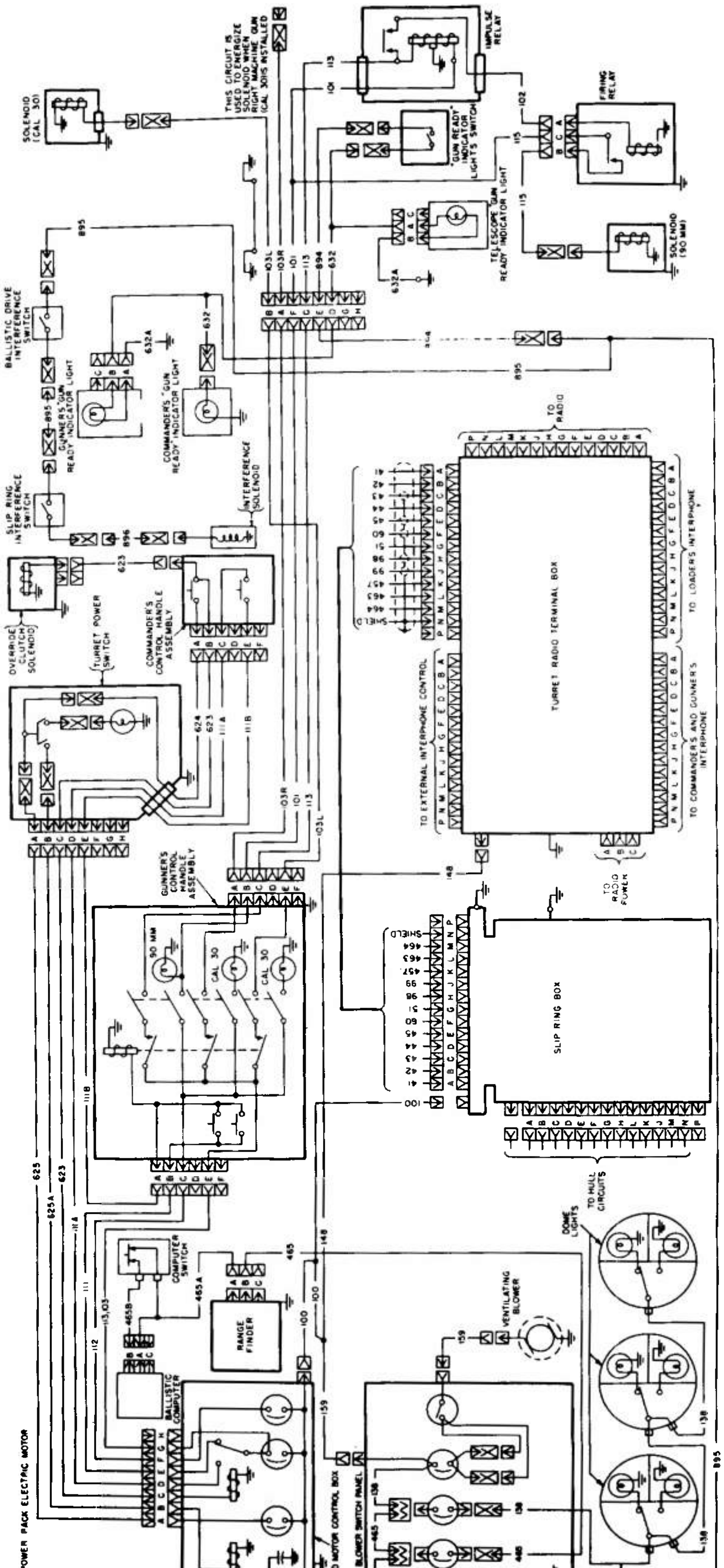


Figure 3-20. Typical Electrical Wiring System for Combat Tank Hull

showing a typical arrangement for interconnecting these systems is shown in Fig. 3-21. Other types of vehicles may have more or fewer electrical subsystems depending upon their missions and special on-board equipment.

Since the turret of a combat tank is not considered to be a part of the hull (par. 1-1.3.2), the turret electrical system is considered as part of the turret. However, just as the turret and hull meet on a common ground—the turret bearing—so the electrical systems of these two major components meet in the turret slipring box (Fig. 3-22). This is located in the center of the turret opening beneath the turret floor. It consists of a lower section, which is secured to the hull by a mounting bracket, and an upper section, which rotates with the turret to which it is connected by means of an outrigger arm. It provides power and interphone connections from the hull to the turret through a system of rotating rings and stationary brushes, and also provides passage of filtered air from the particulate filter kit located in the hull, through an air plenum in the slipring box, to the various electrical components in the turret.

In general, all of the electrical equipment and devices found in military vehicles have certain requirements in common. They require protection from water, weather, dirt, mechanical shock, and accidental short circuiting; some require heating in cold weather for efficient operation; some generate a considerable amount of heat when operating and require cooling; most are, at least potentially, capable of generating sparks and must, therefore, be shielded against explosive or flammable environments; and all apparatus, including the wiring and electrical connectors, must be secured against creating radio interference. At the present time, a 24-v DC system is standard for all military vehicles for all circuits as prescribed by the Department of the Army in SR 705-325-1, *Research and Development of Materiel, Electrical Systems in Motor Vehicles.* Other systems are being considered, primarily a 110-v AC system to be transformed or rectified as needed for various applications. This system has the advantages of being more economical in weight, size, and cost and has a higher efficiency. Its main disadvantage is the difficulty of frequency and voltage control when coupled to the main power plant of the vehicle because this operates at varying speeds and often stands idle for long periods during which the need for electrical power continues.

The uses of electricity on military vehicles fall into eight groups:

a. *Production of mechanical power.* All electrically operated labor-saving and remotely controlled devices, ranging from windshield wiper motors to the traversing motors for large tank turrets and including solenoid-actuated power devices.

b. *Power transmission devices.* Electrical devices that transfer mechanical power from one place to another e.g., electric drive vehicles powered by an on-board generator or electromagnetic clutches.

c. *Storage of energy.* Storage batteries, electrically powered hydraulic and pneumatic accumulators, and various electric hydrospring systems.

d. *Heating.* Devices for heating fuels, lubricants, personnel compartments, and equipment.

e. *Communications.* Radios, radar, television, intercommunications equipment, and remotely operated instruments and controls.

f. *Lighting.* Exterior, interior, and infrared viewing devices.

g. *Engine ignition.*

h. *Firing devices.* For armament.

The power requirements for these uses vary enormously for the many different types and sizes of vehicles. For the smallest transport vehicle, an 18 or 25 amp generator of 430-600 watts is sufficient; for the larger tanks and other combat vehicles as much as 10 to 15 kw may be required. This is usually obtained from two generators, a main and auxiliary. The M103 Heavy Tank, for example, has one main and one auxiliary generator each rated at 300 amp. Load factors also vary greatly on the various vehicles. Some loads, like that of starting an engine, are very high but short in duration; others, like traversing a turret, are high and of longer duration but are intermittent; while still others, like fans, heaters, and pumps, are continuous. A tabulation of some of these loads is given in Ref. 55.

Waterproofing of electrical apparatus on military vehicles is a complex undertaking and should not be conducted in a hasty manner. Although the entire vehicle may be required to survive complete submersion in fresh or salt water, not all parts are necessarily required to

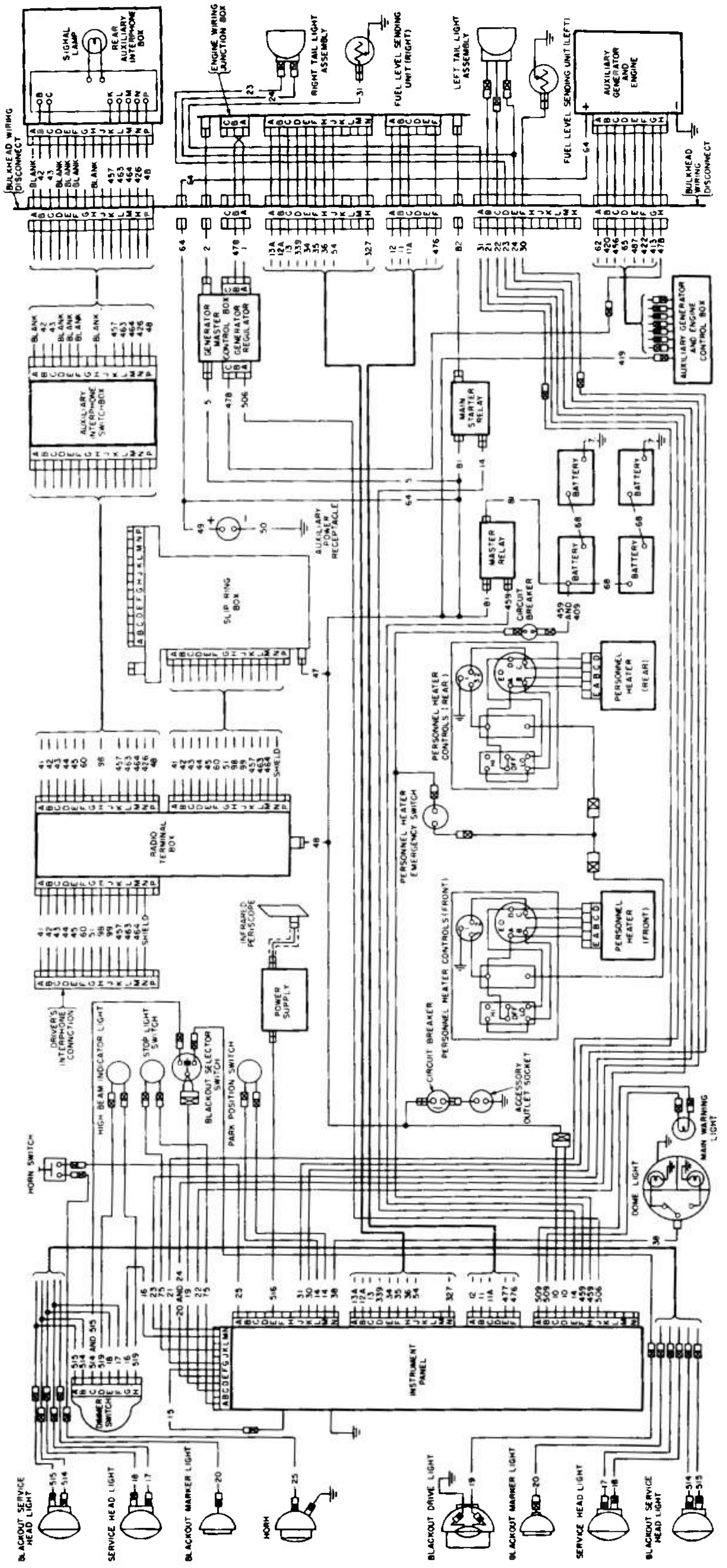


Figure 3-21. Typical Electrical Wiring System for Combat Tank Turret

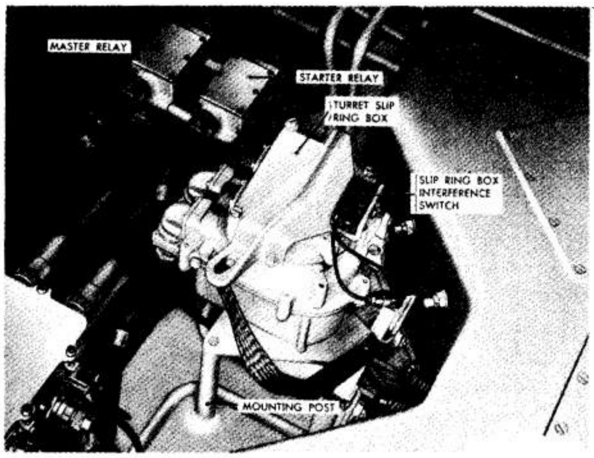


Figure 3-22. Turret Slipring Box—Installed View With Turret Platform Removed

operate while submerged. Components that are completely enclosed in watertight housings generally need no additional waterproofing. These watertight enclosures, however, cut down on heat removal necessary for most electrical devices and create additional hazards. One of these is a breathing problem that results in the condensation of moisture due to sudden temperature changes, particularly when submerged in cold water. This moisture can cause short circuits; corrosion; or, if the moisture freezes on electrical contacts, circuit malfunctions. A further hazard associated with tight enclosures is the possibility of ionization of the atmosphere within the enclosure. This may cause a breakdown of the electrical insulation and lead to subsequent circuit malfunctions and failure.

3-19 GUN TRAVERSING AND ELEVATING SYSTEMS

The basic requirements of gun traversing and elevating systems are reliability, endurance, low power demands, speed, and precise control. Manual traverse and elevation are used extensively in adjusting the final lay of the gun when movements of only a few miles are required. However, the speed of power traverse and elevation is essential to bring the weapon to the point of final lay in a minimum of time. Two basic types of power systems are used, namely, electric and electrically powered hydraulic systems.

Electric systems consist essentially of direct current motors that drive the traversing and elevating mechanisms through appropriate gear trains. Elevation of the gun is sometimes accomplished by means of a hydraulic cylinder which is powered by an electric motor-operated hydraulic pumping unit. All controls, for both traversing and elevating, are electrically operated These systems require prodigious amounts of electrical power, but their redeeming attribute is precise rate control despite variations in friction or load variations caused by rough terrain, abrupt maneuvers, or operations on steep slopes.

Heat dissipation is a major problem associated with electrical equipment. High torque loads placed on electric motors cause increased currents to flow through their windings and results in the production of heat. When operating over rough terrain or when executing abrupt maneuvers, the traversing and elevating mechanisms experience servere shocks. The electric motors are then called upon to instantaneously exert momentary high torques ats very low speeds, a considerable amount of heat is generated. Ventiliation is employed to dissipate this heat, but the operation of the ventiliating equipment also adds to the power drain.

The electric system is very much favored for light and medium tanks—as is the hydraulic system. In the case of heavier tanks, however, where the turret weight attains appreciable proportions, the electric system becomes undesirable because of space requirements. In a large tank, a traversing motor of approximately 2-1/2 hp is necessary. A motor of this size not only drains considerable power, but also occupies considerable space—fighting space—within the vehicle.

Hydraulic systems use electric power to operate motor driven pumps to develop hydraulic power for traversing motors and gun elevating cylinders. A hydraulic accumulator is usually incorporated into the system to maintain a constant hydraulic pressure without the need for continuous operation of the electric motors. The turret can usually be traversed about 80° with the pump motor circuit de-energized.

3-20 STABILIZING EQUIPMENT

In order for combat vehicles to fire their armament effectively while moving over rugged terrain, stabilizing equipment is required to

counteract the effects of vehicle motion on the positions of the gun and sighting telescopes. In effect, the gun and sighting telescopes remain fixed in space while the vehicle pitches, rolls, and slews. Gyroscopes sense the angular movements of the vehicle axes and translate them into appropriate signals to the traversing and elevating mechanisms of the main armament and sighting equipment to maintain these on target. The electrical requirements of a gyrostabilized telescope are not particularly severe. However, the electrical requirements of such a system applied to a heavy turret and gun, especially when operating over rough terrain, may reach 10 to 15 kw.

3-21 MISCELLANEOUS MOTOR-DRIVEN DEVICES

A number of miscellaneous electric motors are employed in military vehicles in addition to those already mentioned. These vary from very small motors, such as used in electric windshield wipers and small fans for cooling low-wattage electronic equipment, to large motors, such as used to drive large hydraulic pumps. Some are continuous-duty motors whereas others operate intermittently. Most of these motors are of the DC type having a commutator and brushes. All are required to possess maximum reliabilitiy for the performance of some function vital to the successful operation of the vehicle. Amphibious vehicles, and certain other vehicles that are equipped with a totally enclosed hull, require bilge pumps to remove water that may have entered during an amphibious or deep-fording operation. These bilge pumps are usually electrically driven pumps that, due to their location, must be extremely rugged and completely waterproof.

3-22 HEATING AND COOLING

The need for supplying heat to various components and compartments of the military vehicle places an additional load upon the electrical system, particularly during cold-weather operations. In addition to the heat required for crew comfort and windshield defrosters, heat must be supplied to various components of the fuel system to prevent ice or frost from clogging minute orifices. Hydraulic systems are heated to prevent sluggish operation; pneumatic systems are heated to prevent the formation of ice; optical systems are heated to prevent fogging or frosting; and storage batteries are heated to maintain their capacity.

Electronic equipment, hydraulic equipment, air compressors, and electric motors all generate heat when operating and create an appreciable cooling problem. These cooling requirements place another load upon the electrical system but, fortunately, not an additional load, because heating and cooling are not required simultaneously by the same piece of equipment.

3-23 COMMUNICATIONS

Two types of communications equipment are found on military vehicles—intercommunication (intercom) systems and outside communication equipment. Intercom systems are usually required in combat vehicles to maintain contact between the crew members while they are manning their combat stations. In the 280 mm gun carriage, an intercom system is used between the drivers at each end of the extremely long vehicle. Certain tactical vehicles, such as the armored personnel carrier, require an intercom system because of the high noise level within the vehicle. In general, intercom systems do not constitute a very heavy electrical load. Radio equipment for communication to stations outside of the vehicle, however, does constitute an appreciable load. It was largely on account of this load that the 24-v system was adopted for military vehicles. This radio equipment includes radio receiving and transmitting equipment, and radar and infrared surveillance equipment.

Another class of communication equipment in military vehicles comprises the multitude of instruments and indicators. The function of most of these instrument systems is to sense various factors that affect the vehicle and its mission, and to present them to the attention of the driver or the responsible crew members. Speeds, temperatures, pressures, operating characteristics of vehicle components, etc., comprise the factors that the instrument systems electrically measure and convey from the sensor to the indicating instruments. Some of these instruments are of the rheostat type, some are of the potentiometer type, and some employ combinations of synchros.

3-24 FIRING OF ARMAMENT

An electrical circuit peculiar to military vehicles is that circuit concerned with the firing

of weapons mounted upon the vehicle. Some guns are fired by electrically heating a thermal detonator; others, by a solenoid-operated percussion mechanism. The current required by these devices varies from about 4 amp for electrically fired machine guns, to 25 amp for some large guns. Safety relays incorporated into the firing circuits prevent accidental firing.

3-25 LIGHTING

The lamps used in military vehicles are standard gas-filled incandescent lamps with tungsten filaments. The voltage of the lamps must correspond to the design voltage of the electrical system unless a resistor is inserted to lower the voltage to the lamps. Control lights are usually of a very low wattage because they primarily indicate a function and do not illuminate an object. Lamps range in size from small 1/2-cp instrument lamps to 50-cp or more driving lights. These lamps must be able to withstand the severe vibration and shocks to which a military vehicle is subjected. All lights on the exterior of military vehicles must be totally waterproof. Because of the low operating voltage of automotive lamps, the current requirement is high. A lamp having two filaments, one of 32 cp and the other of 21 cp, will draw 3.9 and 2.8 amp per filment, respectively. With this current requirement, lighting must be considered seriously in computing the power requirements of an automotive electrical system. The vehicle specification should also be checked as to the lighting requirements of the vehicle and the type of lamps to be supplied. Standard military vehicle lights are described in MS-51302, MS-51303, MS-51318, MS-51319, and MS-51320.

Additional discussion on lights and lighting is given in pars. 3-13 and 3-18.1.

SECTION VII—MISCELLANEOUS DEVICES AND HARDWARE

3-26 ENGINE AND TRANSMISSION MOUNTS

Engine and transmission mounts support and maintain the position of the power plant in the vehicle during all static and dynamic conditions. Their specific functions are (1) to provide sufficient support points to distribute the engine and transmission loads over the supporting structure, (2) to contain the power plant torque developed by rapid accelerations and high speeds, and (3) to isolate the power plant from shock loads resulting from travel over rough terrain and from ballistic impacts and blast. The degree of rigidity, number of mounting points, and type and amount of shock attenuation required depend upon a number of variables—such as the type and mission of vehicle; power plant size, weight, and location within the vehicle; vibrational characteristics of the vehicle's suspension system and frame; and specifications of acceptable noise and vibration limits—and must, therefore, be treated on an individual vehicle design basis. Generally, the power plant mounting supports are in the form of machined pads that are welded directly to the hull. Exceptional cases exist where the hull is too thin to support the loads or where power plant access is required. In these cases, a framelike structure is employed to support the power plant, and this is bolted or welded to load-bearing points on the hull.

3-27 SUSPENSION MOUNTS

The functions of the suspension mounts are to provide attachment points for the suspension linkage, to transmit residual suspension system loads into the hull or frame of the vehicle, and to transmit vehicle loads and shocks originating on the vehicle—such as weapon recoil forces and blast loads—to the suspension system. Where the hull configuration is such that no frame is required, the suspension mounts are usually welded directly to the hull and are placed adjacent to the wheel or bogie positions. Where the hull construction is such that a frame is required, however, the wheel or bogie positions usually do not coincide with the most efficient load bearing points on the frame. In such cases a suspension support structure is needed to distribute the suspension loads to the frame. One such system[58] uses a torque tube on each side of the hull extending for the length of the frame. These torque tubes are welded to transverse frames on the bottom of the hull, and

the suspension mounts are welded to the tubes. Regardless of the suspension mounting techniques used, care must be exercised to minimize the intrusion of the suspension mountings into the road clearance. The bottom of the hull should be as smooth as possible to minimize bottom drag. For additional discussions on suspension mounts see par. 2-12.2 and Ref. 59.

3-28 SHOCK MOUNTS

Military vehicles are subjected to a variety of shock conditions of various levels of intensity. In addition to the normal shocks associated with vehicular motion, the modern military vehicle is subjected to combat-induced shocks and shock loadings associated with airborne delivery. Shock mounts should be provided for all shock-sensitive equipment to prevent its damage and also for all devices whose efficient employment would be degraded by shock or servere vibrations. This applies particularly to such devices as fire control instruments, optical devices, and electronic devices. In addition, thick, soft rubber cushions are usually provided around the viewing ends of optical devices (view blocks, periscopes, telescopes, range finders, etc.) to protect their users from injury and to serve as steady-rests for their heads.

For a comprehensive treatment on the design of shock mounts and vibration isolators, see Ref. 60.

3-29 BULKHEADS

Bulkheads are vertical partitions in a vehicle hull which separate the various compartments. They have several functions—such as providing additional ballistic protection to specific areas, imparting rigidity and strength to the hull structure, and isolating specific vehicle areas (crew, cargo, power plant, ammunition storage) to protect them from fire, fumes, heat, noise, etc. A bulkhead sometimes is requred to provide ballistic protection in applications where the external hull thinly covers a nonvulnerable vehicle area—e.g., a bulkhead between a cargo compartment and the crew compartment.

An overall vehicle weight saving can often be achieved by using reinforcing bulkheads to add strength and stiffness to a hull rather than by increasing the thickness of the hull plates. Bulkheads provide strength and rigidity to hulls by functioning as local strong points at which loads can be distributed to the hull structure, and by serving as shear members to stiffen hulls under torsional loads.

The use of bulkheads for isolation of compartments has many applications. Some typical examples are isolating the engine from the crew compartment to reduce the transfer of engine noise and fumes; isolating the crew from the cargo compartment to protect the crew from shifting cargo, to reduce the heating and ventilating loads, and to improve ballistic and CBR protection; isolating on-board ammunition from the heat of the engine and protecting it from accidental damage; fire barriers to reduce the effects of a fire in the vehicle; and the formation of flotation chambers in amphibious vehicles.

3-30 FIRE EXTINGUISHING SYSTEMS

Two general types of fire extinguishing systems are used in military vehicles—portable and fixed. *Portable extinguishers* are hand-held devices provided to combat fires that may occur in the driver's or fighting compartments of the vehicle. They usually consist of a metal cylindrical pressure tank filled with carbon dioxide (CO_2) and equipped with a discharge nozzle and an actuating trigger. Mounting provisions should be made in a readily accessible location of the crew compartment for one such extinguisher. The mounting should permit quick removal of the extinguisher. Other materials, such as chemical foams and dry materials, are also used in portable extinguishers; but CO_2 is the most commonly used in military vehicles because it is clean, dry, noncorrosive, effective on all types of fires, and doesn't damage materials that it contacts. It is particularly effective in an enclosed space, because its action is to dilute and displace the oxygen in the vicinity of the flames until the air can no longer support combusion. Liquid CO_2 can be stored in steel cylinders under high pressure and, in this condition, has an expansion ratio of 450 to 1. This allows a large volumetric coverage with a reasonably compact extinguisher. This also makes the pressurized cylinders potentially very dangerous—even when only partially full. Therefore, appropriate safeguards should be established when a CO_2 cylinder is removed

from the system or when CO_2 lines must be disconnected for servicing, maintenance, or repair.

Fixed fire extinguishing systems consist of one or more CO_2 cylinders semipermanently mounted in a readily accessible location in the vehicle, usually in the driver's compartment. Rigid metal tubing connects these cylinders to discharge nozzles located in areas that are vulnerable to fires, in areas where fires are likely to occur, or in areas that are normally unattended or inaccessible during operations. Engine compartments, particularly engine compartments of combat tanks, are typical areas normally protected by fixed fire extinguishing systems. The M60 Tank has three fixed CO_2 cylinders, of 10-lb capacity each, located in the driver's compartment. The system can be operated from either the inside or outside of the vehicle. An exterior control handle is provided on top of the hull for this purpose. All three cylinders discharge simultaneously when the system is actuated (Fig. 3-23).

3-31 FENDERS AND SHROUDS

3-31.1 FENDERS

Fenders are provided over the wheels and tracks of most military vehicles. Separate fenders are usually not necessary over the rear wheels of vehicles whose bodies overhang the wheels nor over wheels that are recessed in wheel wells. Similarly, they are omitted from the rear wheels of truck-tractors (Fig. 1-13) when the overhang of the coupled semitrailer will make fenders unnecessary; although they are sometimes still included to provide a covering for the wheels when traveling without a semitrailer or when towing a trailer that does not overhang the truck tractor wheels. Fenders are also usually omitted from certain slow moving engineer vehicles, such as large tractor-dozers, crawler-type crane/shovel vehicles, etc.

The principal function of fenders is to intercept the mud, water, and stones that get picked up and hurled into the air by the rotating

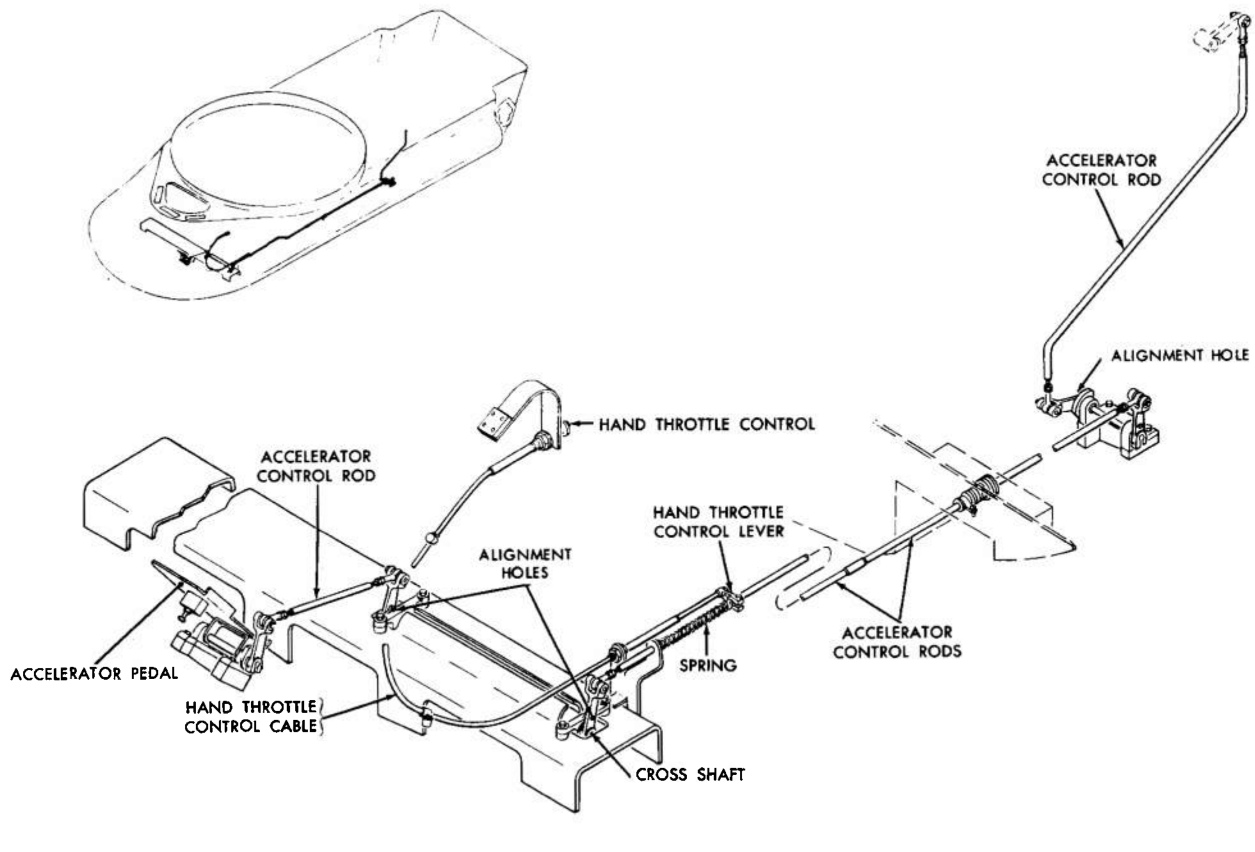


Figure 3-23. Schematic Diagram of a Fixed Fire Extinguisher Installation in a Combat Tank

wheels and tracks and deflect them downward so as not to interfere with the driver's vision, create a hazard for following vehicles, or unnecessarily splatter the vehicle with mud and other material. On amphibious vehicles that depend upon their wheels or tracks for propulsion in the water, fenders have an additional function—they redirect the water that is carried around by the rotating wheels or tracks to develop thrust in the desired direction. This is discussed in more detail later in this pargraph. Fenders on tracked vehicles also serve to prevent dirt, sand, and rocks from being thrown by the tracks onto the vehicle's top deck where an accumulation of such debris over the intake grilles can impede the ventilating system, interfere with vision, and even degrade engine performance. In addition, fenders provide a degree of protection from the moving tracks to personnel who sometimes ride on the outside of tracked vehicles.

Fender design considerations are primarily a matter of clearances. Fenders for wheeled vehicles should be sufficiently wide to accommodate dual wheels when these are specified for the vehicle. Adequate clearance should be provided between wheels and fenders to allow the full excursion of the suspension system and to permit the installation of traction devices when these are necessary. Additional lateral clearance is usually required around the front wheels to allow for steering. The under surfaces of fenders should be reasonably smooth and free of obstructions, ledges, and pockets that can cause a build up of mud, snow, and foreign debris and make cleaning difficult.

The widths of most wheeled vehicle bodies—particularly cargo and van bodies(Figs. 1-6, 1-7, 1-9, and 1-11)—generally extend well beyond the outer faces of the rear wheels in order to attain maximum cargo space. By allowing that portion of the body which overhangs the wheels to perform the function of fenders, no separate fenders are required. In these cases, vertical deflectors are usually installed ahead and behind the wheels to prevent the throwing of debris forward and to the rear. These should not extend much below the centerline of the wheel or they may reduce the vehicle's maximum angles of approach and departure.

Fenders on tracked vehicles equipped with track support rollers (Fig. 3-24) do not require as much clearance above the track as do fenders on wheeled vehicles or on vehicles with flat track suspensions (Figs. 1-28 and 1-29) because of the differences between the operating characteristics of their respective suspension systems. In wheeled and flat tracked vehicles, the actions of the suspension systems result in a constantly varying distance between the top of the wheels or tracks and the bottom surfaces of the fenders. When track support rollers are used, however, the upper portion of the track is held at a relatively constant distance with respect to the fender (Fig. 3-24), and only the lower portion of the track moves vertically as a result of the elastic action of the suspension system. Thus, only sufficient clearance is required above the track to minimize the buildup of foreign

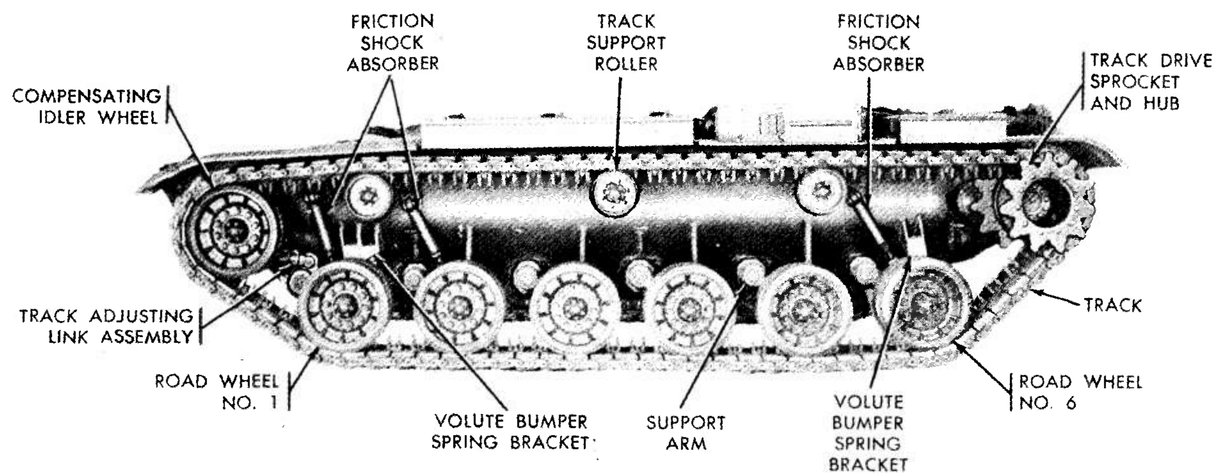


Figure 3-24. Suspension System of M60A1 Combat Tank Showing Relationship of Fenders to Track

debris, provide access to components, and to clear any up and down whipping of the track that may occur at high speeds. Generally, fenders on tracked vehicles are designed to be easily removed to provide access for maintenance and repair. Stowage boxes are often incorporated onto their upper surfaces to utilize this area.

Figs. 3-25 and 3-26 illustrate the fender configuration of the M60A1 Combat Tank. The fenders and their stowage boxes are constructed of aluminum alloy. They are bolted to steel outriggers which are, in turn, bolted to bracket pads welded to the hull. This construction permits the removal of the fenders and stowage boxes when this is necessary to reduce the vehicle's overall width for transporting or when damaged components require repair or replacement.

Certain amphibious track-laying vehicles propel themselves through water by the action of their tracks in a manner similar to that of early paddlewheel boats. On early tracked ampibians (1940-1945), the upper or returning portions of the tracks traveled entirely out of the water and the vehicles achieved fairly good water speeds (7.5-8.5 mph). Current vehicles of this type (Fig. 1-25), because of their greater weight and lower track envelopes, float with their entire tracks submerged. Without properly designed fenders and shrouding, a totally submerged track will develop practically no useful thrust because the thrusts from the upper and lower track strands cancel each other. Therefore, fenders and shrouds on these vehicles perform the additional function of channeling the water accelerated by the upper track strands and redirecting it so that it is discharged from the return track in a direction opposite to the direction of travel. Experience with track-propelled amphibious vehicles has developed a number of factors that influence their speed in water[61-64]. These are presented here to serve as guidelines in the design of this type of vehicle.

a. Front fenders should be carried downward around the front sprockets as far as is practicable—up to 150° from the point of tangency of the track—for improved thrust and speed. Extending the fenders beyond this 150° point has very little, if any, additional effect. Any additional thrust that is developed by

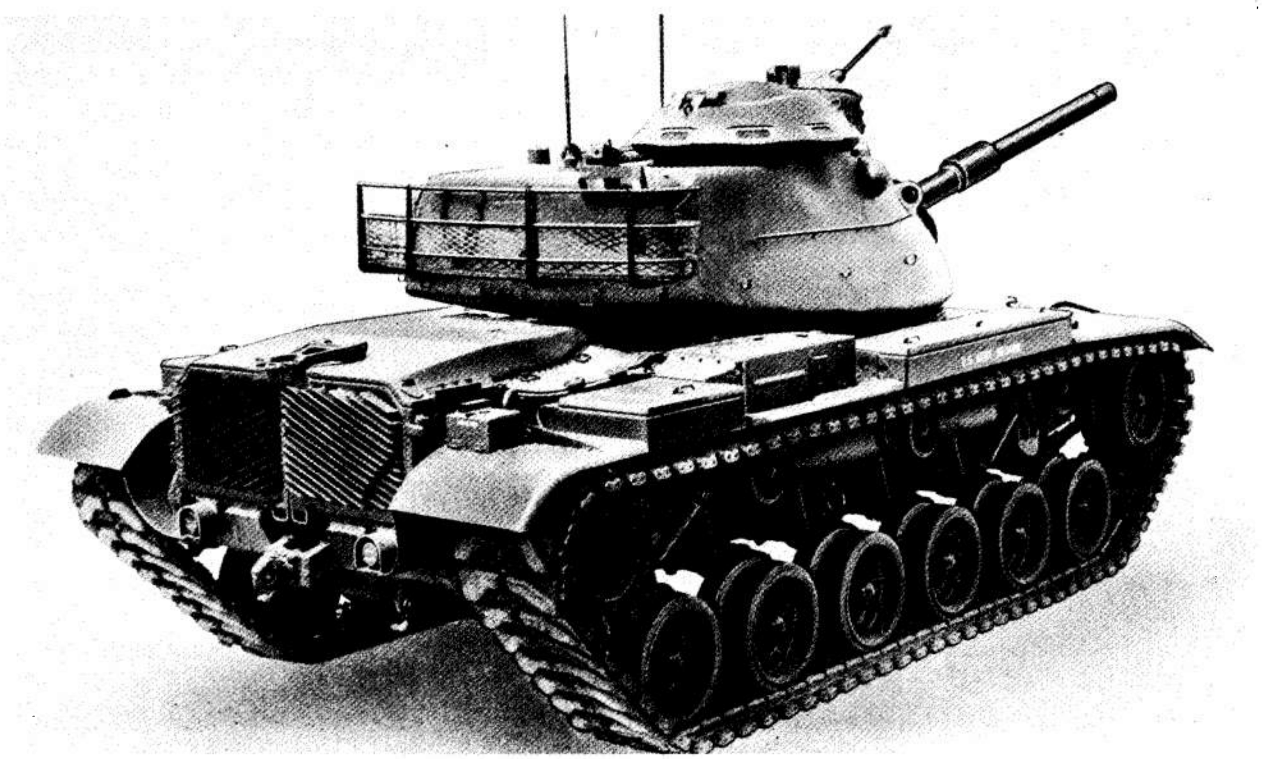


Figure 3-25. Right Rear View of M60A1 Combat Tank

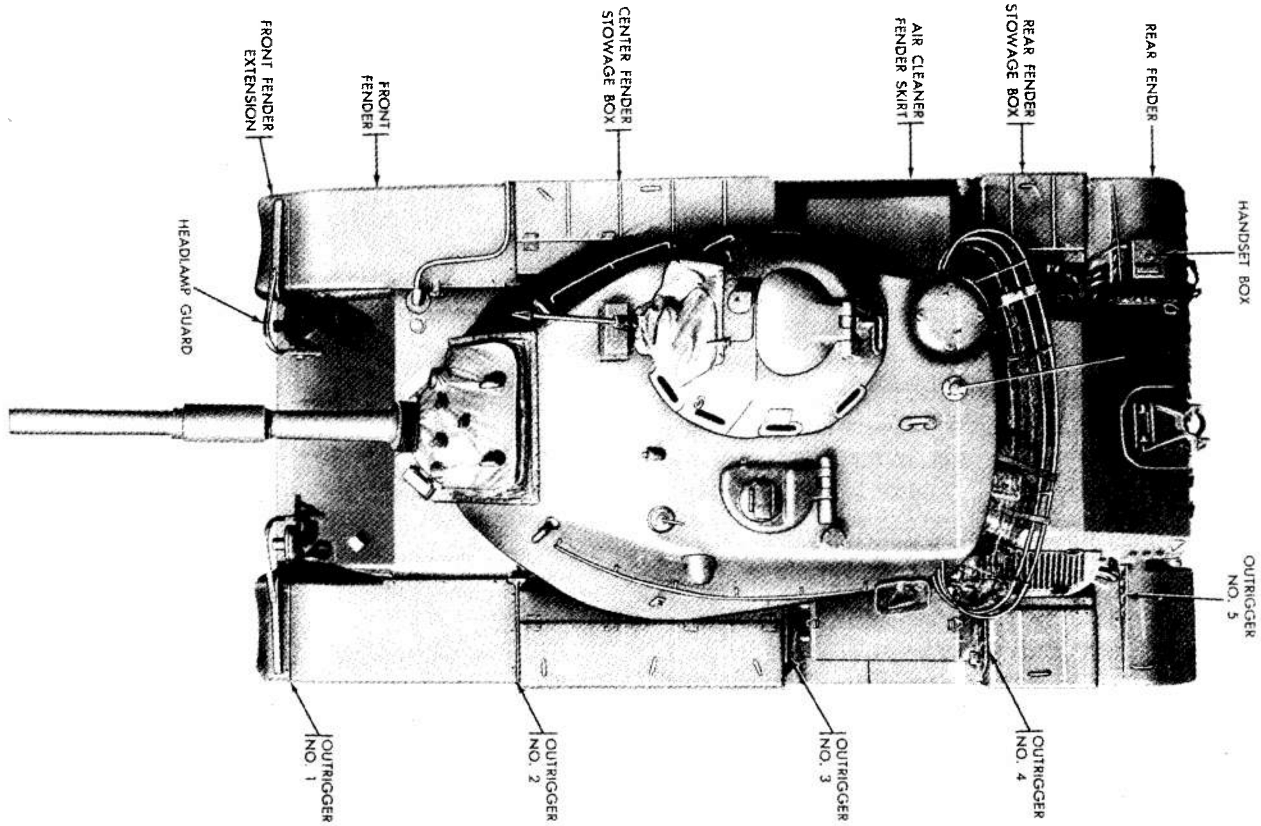


Figure 3-26. Top View of M60A1 Combat Tank Showing Location of Fenders, Outriggers, and Fender Stowage Boxes

extending the fenders beyond this point seems to be cancelled by the increased drag resistance of the extension. Other design considerations—such as the profile of the track envelope, specified angles of approach and departure, location of the front sprocket with respect to the forward edge of hull, and restrictions on overall vehicle length—may tax the designer's ingenuity to attain a 150° fender wrap-around. Hydraulically operated retractable front fenders have been designed to meet these requirements. One such design is described in Ref. 61. Wrap-around front fenders on propeller driven amphibious vehicles are a detriment as they increase the frontal drag resistance and serve no useful purpose in developing thrust.

b. Rear fenders—extending around the rear sprocket to a maximum of 150°—are required for effective water speeds in reverse. A tracked vehicle with wrap-around front fenders but no rear fenders will continue to move in the forward direction (though with very little thrust) even after the track rotation is reversed. This is due to the front fenders deflecting the water being carried upward by the reverse rotating lower track strand and thus creating a thrust in the forward direction. Stern baffles, plates between the sidewalls of the hull and the shrouds to close off the entry of water to space above the return track, are effective in improving thrust.

c. Stern planes and contravanes have a marked influence on the propulsive efficiency of tracks. Stern planes are broad plates, one for each track, that extend horizontally rearward from the transom and laterally to the outside of the tracks. Their function is to deflect the stream from the tracks horizontally and thus increase the forward thrust. Contravanes are discussed in Ref. 65. These are similar to stern planes except that their elevation, angle of attack, and size—which are dependent upon the initial trim

of the craft—are optimizied. The force of the water against the vanes can raise the stern enough at high speeds to cause the vehicle to trim down by the bow. Since no single position or angle of attack is optimum for all trim conditions, the attainment of maximum benefit demands that the contravanes be designed with some adjustability. In all cases, for maximum efficiency, the rear sprocket should be closed off by a stern baffle.

3-31.2 SHROUDS

Shrouds are skirtlike attachments fastened to the outboard edges of the fenders and extending downward to conceal the upper portion of the track and suspension system (Figs. 1-25 and 1-31). Their principal function is to improve the propulsion efficiency of the tracks in water. It does this by helping to direct the water that is carried forward by the returning track strand toward the rear to develop an additional forward thrust. For maximum effectiveness, minimum lateral clearance should be provided between the shroud and the track commensurate with freedom from mud impaction during operations in mud.

One concept[62] utilized the water entrained around the top of the track by the shroud to improve the vehicle's steering control in water while also improving its water speed. It accomplished this by incorporating a series of parallel vertical cupped vanes in the forward portions of the track shrouds. The vanes intercepted some of the water being driven forward by the upper track and reversed its direction toward the rear as they discharged it. In this manner, water speed was increased by 13.5 percent, and steering was also improved because the water was exhausted at an angle to the side of the vehicle.

3-32 CONTROLS

Controls normally associated with the hull are mechanisms that regulate the starting, running, steering, and stopping of the vehicle or that regulate or monitor the operation of special equipment mounted on the vehicle. The controls usually found on a military vehicle are steering wheel and related linkage, hand- and foot-operated brakes, hand- and foot-operated throttle controls, transmission shifting lever and associated linkage, pivot steering selector (if so equipped), engine start and shut-off controls, and controls for special equipment such as weapons, turrets, dozer blades, scraper blades, booms, winches, and control linkages necessary to convert the vehicle from land to water operations.

Design objectives of control elements are positive vehicle and equipment control, a high degree of operator efficiency, simplicity of the control function, safety, ease of maintenance, and a long, trouble-free life. Control systems comprise any of the following elements:

a. Flexible cable-operated force or displacement transferring systems

b. Mechanical linkages

c. Hydraulic systems

d. Pneumatic systems

e. Electrical circuits

f. Electromechanical or electrohydraulic servosystems

g. Power relay amplifiers

The combination of elements selected for a particular control function depends upon the magnitude of the forces or displacements required, the required precision of settings, allowable response times, the distance over which control signals must be transmitted, vulnerability to spurious signals, vulnerability to physical damage, and considerations of reliability, compatibility, practicability, and economics. Automatic controls are generally provided with a manual over-ride to permit manual correction or manual control in the event of a power failure. Similarly, power-boosted devices, such as power steering or power brakes, are designed to be operated manually in the event of a power failure.

The subject of controls is beyond the scope of this handbook. It concerns the body or hull designer insofar as he must provide for the mounting, housing, and protection necessary for the various control components and for the passage of control elements, such as cables and linkages, through bulkheads, walls, floors, and ceilings. This necessitates considerations of seals and sealing which are discussed in pars. 3-7 - 3-9.

3-33 CBR EQUIPMENT

Chemical, biological, and radiological (CBR) equipment that might be found on a military vehicle is of two general types—(1) devices for

the dissemination of CBR agents, and (2) defensive devices for use against these agents. The dissemination of CBR agents is largely a matter of selecting the proper ammunition, and the actual dissemination is done by means of the primary weapon with which the vehicle is armed (cannon, mortar, rocket launchers, etc.). Defensive devices comprise such items as personnel gas masks, protective clothing, decontaminating equipment, and high performance particulate air filtering systems.

It is contemplated that future military vehicles may be constructed with completely sealed personnel compartments equipped with closed cycle life support systems. The weapons, power plant, and related equipment will be external to the personnel compartment and will be controlled from within by means of remote control devices. An important consideration when designing a vehicle of this type is that of egress of personnel after the vehicle has been under CBR attack, or has been contaminated by exposure to CBR materials, and has returned to an uncontaminated area. Under these circumstances, the vehicles's hatches or doors will be contaminated. Unless precautionary measures are taken when the hatch is opened, contaminants may enter the compartment. Contaminants may also be picked up when exiting personnel contact exterior surfaces of the door or hatch cover. This can be prevented by designing the hatchway with a retractable liner to provide a barrier between the personnel and the contaminated exterior surfaces. The general subject of CBR warfare and the considerations it imposes upon the vehicle designer are discussed in par. 2-6.3.3.

3-34 PRIMARY WEAPONS

Combat vehicles are generally equipped with one primary weapon—the remaining weapons are designated as secondary. The primary weapon is that weapon which enables the vehicle to accomplish its primary mission and may be a cannon (gun, howitzer, mortar), a type of small arm (generally less than 1-inch bore), a missile or rocket system, or some type of CBR projector. It is generally mounted in a turret or cupola to facilitate traversing the weapon, although this is not always the case. Mortars, being high angle of fire weapons, are generally mounted on the hull floor. The roofs of mortar-mounting vehicles are made either retractable or are left open to permit firing the weapon from the vehicle. In these cases, the floor structure must be made sufficiently strong to withstand the firing loads. Relationships between the primary weapon and the hull and turret that are of concern to the hull designer are discussed in pars. 2-12.3 and 2-12.4.

3-35 SECONDARY WEAPONS

The secondary weapons of a combat vehicle are all of the weapons with which it is equipped with the exception of the primary weapon. Their purpose is for antiaircraft defense, defense against infantry, and for the accomplishment of such secondary missions as attacking infantry positions, supporting infantry advances with automatic weapon fire, neutralizing enemy machine gun positions, and for counterambush operations.

Secondary weapons consist of small caliber (under 1-inch) automatic weapons, rocket launchers, flame throwers, grenade launchers, and smoke or CBR projectors. Not all of these are found on any one vehicle, of course; but various combinations of these are used. The secondary armament of the XM551 Armored Reconnaissance/Airborne Assault Vehicle, for example, consists of a cal .50 heavy barreled machine gun on an all-purpose mount, a 7.62 mm machine gun mounted coaxially with the primary weapon (a 152 mm combination gun and missile launcher), 8 grenade projectors (4 on each side) for counterambush, and a cal .45 hand-held submachine gun for close-in defense.

Secondary weapons may be mounted in the main turret, in a cupola atop the hull (Figs. 1-23 and 1-25), directly in the hull, or at any combination of these locations. They may be mounted coaxially with the primary weapon, they may be pedestal-mounted, mounted in the bow, mounted in pods at convenient locations on the vehicle, or they may be turret-mounted in a rotatable cupola atop the main turret or in any other suitable location (Fig. 3-27).

Coaxially mounted guns have limited elevation and depression, since these movements are limited by the movement of the primary armament. Futhermore, the opening required for this type of mount increases the vulnerability of the gun shield, and crowded conditions in the turret often make loading and servicing the coaxial machine gun slow and

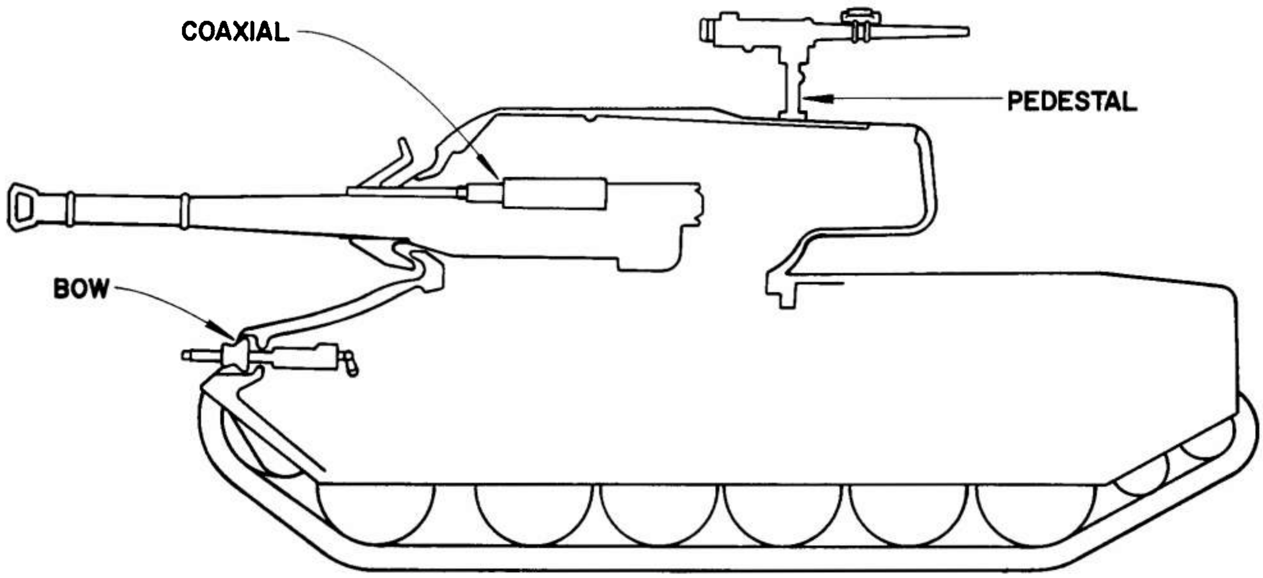


Figure 3-27. Standard Machine Gun Mounts

difficult.

Pedestal-mounted guns are "free" guns which makes them essentially hand-held weapons. Aiming of pedestal-mounted guns is done by means of the same iron sights found on ground-mounted machine guns and by tracer control. Elevation is limited by the pedestal height, which is usually made low to decrease the vehicle's silhouette. A pedestal mount on top of a turret interferes with sighting through the vision blocks in the commander's cupola. Another disadvantage of this type of mount is that it exposes the gunner to enemy fire. Weapon traverse is limited by the length of the gunner's arms and the amount of his body he is willing to expose. Full 360° traverse is obtainable only by climbing out of the turret and "walking" the gun around or by traversing the turret.

Bow mounted secondary weapons necessitate an opening in the frontal armor plate thus increasing the vulnerability of the vehicle. Futhermore, the use of sights on a bow mounted machine gun is impractical; therefore, the gunner must rely on tracer control. Elevation and traverse of the weapon are limited by the configuration of the hull and the available space inside. Current combat tanks have eliminated the bow gun.

Fully rotatable turret-type mounts are perhaps the most effective (Figs. 3-13, 3-14, and 3-26). These provide a rotating section that incorporates a hatch and a weapon mount. Some of the latest versions resemble a miniature turret that rotates on a ring fitted in an opening in the main turret roof or in the roof of the hull. Ammunition is fed to the weapon from the interior of the main turret, or from the hull, through a flexible ammunition chute. Thus, the weapon can be fired while the cupola is buttoned up. Optical sights and vision blocks provide overlapping areas of vision.

Remotely positioned and remotely controlled weapons require a charging system for the chambering of the first round and for the ejection of misfires. In addition, a remote sighting system is required; or the direction of fire can be controlled by monitoring the tracer path with respect to the target and adjusting the weapon accordingly. Pars. 2-12.3 and 2-12.4 discuss some additional relationships between the weapons and the vehicle hull and turret that are of concern to the designer.

3-36 KITS

3-36.1 GENERAL

Kits, in the sense used here, are accessory sets of related components designed to be attached to existing vehicles to give them some particular capability beyond that of the original design.

Such a capability may be required on all vehicles involved in a specific military operation, or it may be required permanently on selected vehicles. For example, all vehicles of a particular division might need to be equipped with fording kits when river crossings are anticipated in the course of a campaign. On the other hand, it might be decided that machine gun mounts are desired on every fourth amphibious truck, or that tank dozers should be distributed one to a company, or truck-mounted A-frames should be allotted at the rate of two per company. To permit this flexibility, selected vehicles are modified by attaching the necessary equipment from appropriate kits. An important consideration, however, is that the basic vehicle design must be compatible to the expeditious mounting and dismounting of this kit equipment without the need for major changes to the basic vehicle.

Vehicular kits can conceivably be used to facilitate the performance of any function or operation by, or in, a vehicle. The most common kits are fording, floating, and swimming equipment; winterizing and desertizing equipment; bulldozing equipment; weapon mounts; winches; A-frames; radios and various electronic equipment; flame throwers; protective devices; and infrared equipment. This list is by no means complete. The kits that are required for a particular vehicle are generally specified in the procuring documents.

Kits should be so designed that they can be easily and quickly installed—generally under field conditions—using standard tools. When the use of special tools, jigs, or fixtures is necessary, these should be supplied with the kit.

Since a kit, by definition, is a vehicle adjunct intended to extend the capability of the equipment, it should not detract from the vehicle's normal functions. Some exceptions to this, however, are sometimes necessary as, for example, is the case with flotation kits. The added weight and dimensional changes resulting from the installation of a flotation kit are acceptable if the degradation is of a reasonably low amount and, particularily, if it is temporary and of short duration. Similarly, the kit should be removable without detracting from the serviceability of the original vehicle.

The service life, or durability, of the kit should be commensurate with that of the basic vehicle. In this respect, however, allowances should be made for the fact that the kit may not be in full time use. Thus, a tankdozer has a prescribed operating life of 5,000 miles, but the required life of the bulldozer kit is only 200 miles or 100 hours of operation[66]. When mileage goals for equipment have not been established by the procuring document or some recognized official directive, practical guidelines toward this end are given in Refs. 66 and 67.

3-36.2 FORDING KITS[68]

The ability to operate in reasonable depths of water greatly enchances the mobility of tactical equipment; hence, all tactical vehicles must meet established fording requirements (see par. 2-2.1.6). Current requirements make a distinction between shallow and deep fording. The first is applied to standard tactical vehicles operating without the addition of special kits (although they may have factory-installed items such as intake and exhaust extensions and waterproof ignition systems). The basic vehicle must be capable of fording a specified depth of water without any special preparation.

Deepwater fording, on the other hand, implies the use of special equipment, usually installed in the field by the vehicle's crew prior to the fording operation. The deepwater fording kit may interfere to some extent with the normal functioning of the vehicle on land and should, therefore, be easily and quickly removable immediately after use. A fording kit is customarily a design requirement for all combat and tactical vehicles intended for amphibious assaults in water depths that approach their overall height.

3-36.2.1 Shallow-water Fording

The waterproofing of a hull-type vehicle for shallow-water fording is a comparatively simple matter. To prevent the entrance of dirt and the loss of lubricant during regular operation, grease seals are provided for bearings in the road wheels, idler wheels, track support rollers, and final drive shafts. These same grease seals are adequate to prevent the entrance of water during fording. Similarly, gaskets and seals normally provided in the final drive housings and torsion bar support housings suffice to exclude water during fording.

Drain valves and escape hatches in the bottom

of the hull must be substantially self-sealing to prevent the entrance of water during shallow-water fording. Openings subject only to splash during shallow-water fording are adequately protected by the sealing normally provided to exclude water during rainy weather operation. Drain values can be opened from inside the tank after fording to drain any accumulated water.

3-36.2.2 Deepwater Fording

To meet deepwater fording requirements necessitates more sealing and ducting than can be economically provided on standard vehicles at the time of manufacture. Therefore, water fording kits (Fig. 3-28) are installed on the vehicles before fording. Although various types of vehicles differ somewhat in the preparation required before deepwater fording, there is enough similarity to regard the requirements and design and preparation factors, described in the paragraphs which follow, as common to all.

The component parts of a deepwater fording kit (Fig. 3-29) should be simple and inexpensive. Immediately upon completion of fording, the vehicle must regain its original firepower and mobility. This postfording requirement dictates that all parts of the deepwater fording kit that interfere with firepower or mobility be jettisonable. Because these parts may not be salvaged for reuse, economy is an important requirement. A kit should be simple to install because tactical situations often limit the time available for preparation.

Deepwater fording with World War II tanks required provision of stacks and necessary adapters for engine cooling air. The space requirements of these components were a disadvantage of the vehicle as well as in the kits. Field installation required considerable preparatory welding and sealing. On postwar tanks, the need for engine cooling air stacks for fording operations has been eliminated by the adoption of air-cooled engines with cooling fans located horizontally on top of the engine compartment. Clutches have been provided to automatically disengage the fan drives when the fan blades encounter the resistance of flooding water. Incorporation of adequate waterproofing of all components, as well as a sealed bulkhead between engine compartment and crew compartment, permits complete submergence of the engine and all components during fording operations. The adoption of air-cooled engines capable of operating completely submerged has eliminated considerable bulk from deepwater fording kits and has reduced field installation work. However, certain venting, sealing, and ducting problems still remain. These problems involve the exhaust, fuel, and electrical systems, and miscellaneous openings.

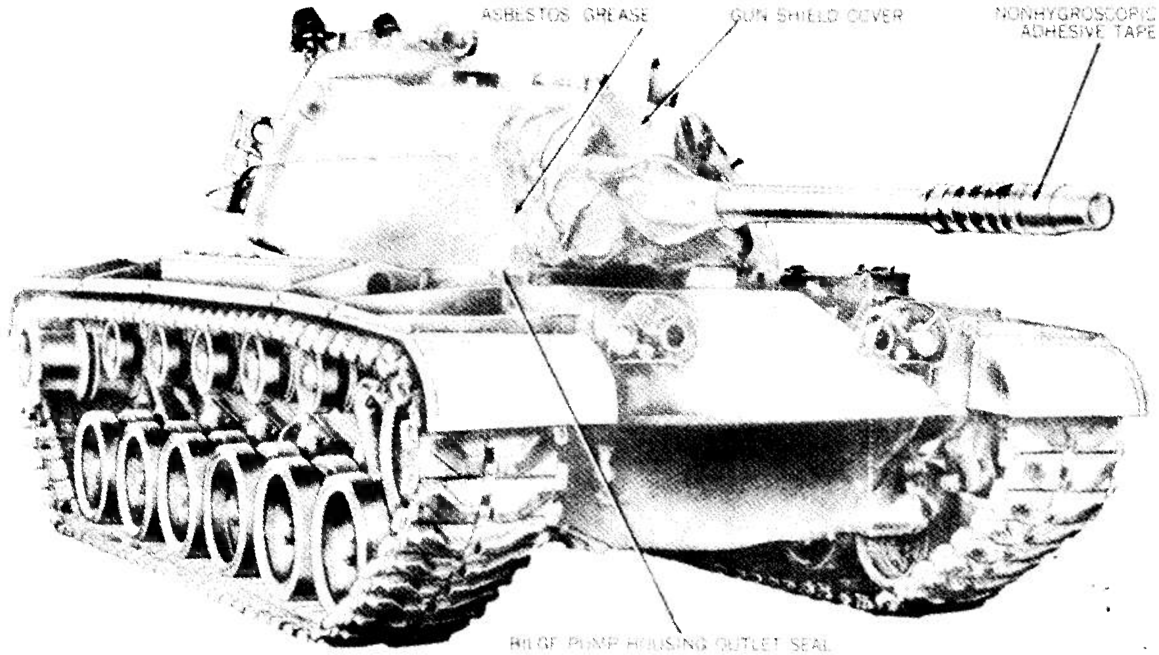
3-36.2.3 Venting for Deepwater Fording

3-36.2.3.1 Final Drive Breather Vents

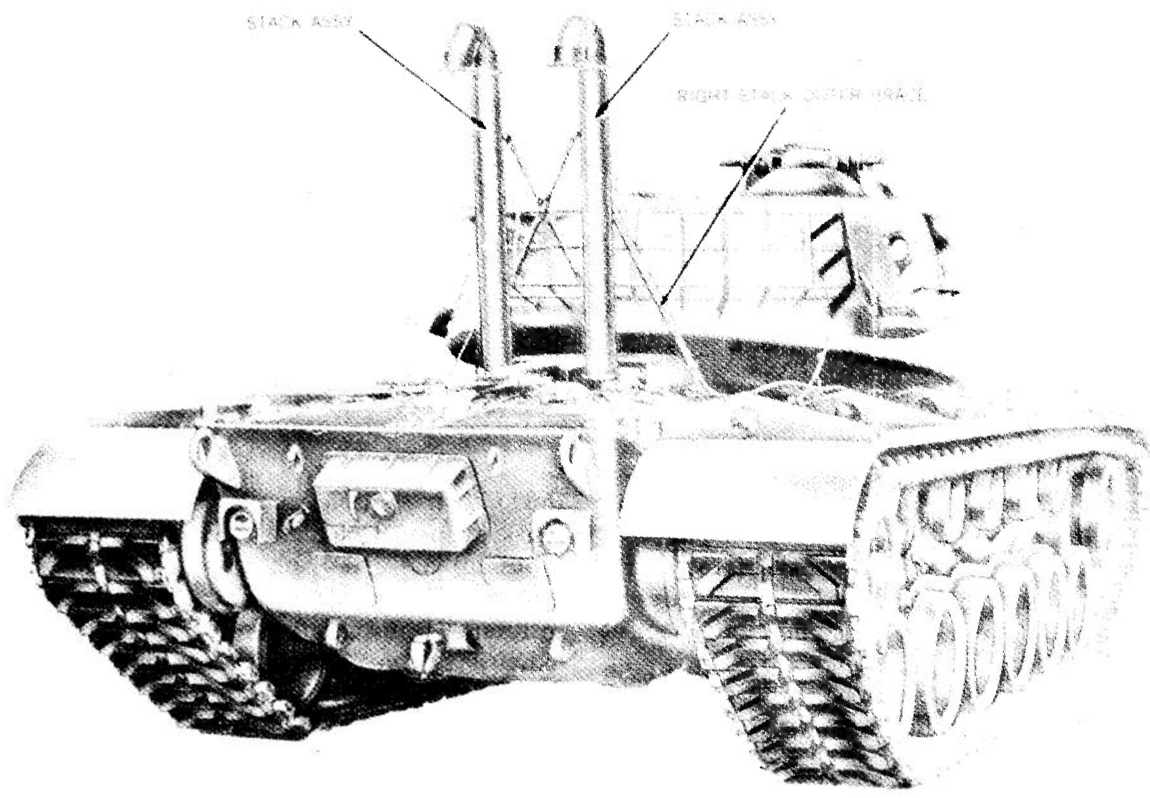
Final drive breather vents are below the water level normally encountered while fording. Water must not be permitted to enter final drive assemblies, yet the vents cannot be sealed prior to fording. If the vents are sealed, sudden cooling of the final drive housing upon submersion creates a temporary partial vacuum within the housing. The resultant pressure difference at the shaft seals could cause serious leakage of water into the housing. Therefore, some provision must be made to vent the final drive assemblies to the atmosphere. Usually, the simplest way to accomplish this venting is to utilize tubing to extend the vents to the crew compartment.

3-36.2.3.2 Exhaust Stacks

One or more exhaust stacks must be provided to conduct the engine exhaust gases above the water level. The cross-sectional area of the stack(s) is determined by engine requirements. Sufficient cross-sectional area must be furnished to avoid back pressure in the exhaust system that would reduce engine power output. On some vehicles the joint seals in the exhaust system require modification before deepwater fording. Some types of joints are entirely adequate in normal operation, but may be a source of serious leaks in deepwater fording. While the engine is running, sufficient exhaust gas pressure is maintained in the exhaust system to prevent water from entering leaky joints. Should the engine stall while submerged, immediate restarting may be possible, provided no other difficulties exist. However, if the engine cannot be restarted immediately, the

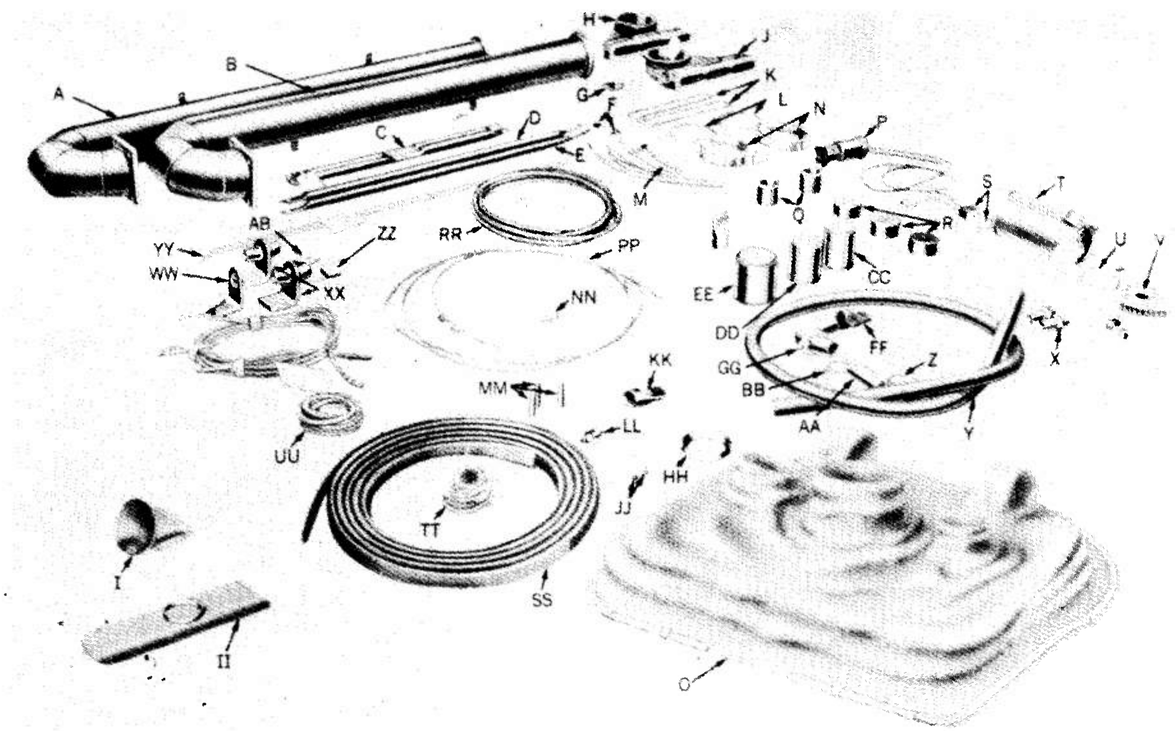


(A) Right Front View



(B) Right Rear View

Figure 3-28. 90 mm Gun Tank Prepared for Deepwater Fording



A	Right stack assembly	I	Collar cap assembly	R	Fire extinguisher horn cap
AA	Nipple	II	Machine gun cover assembly	RR	Fuel tank vent hose
AB	Right stack outer brace anchor	J	Right duct assembly	S	Switch assembly
B	Left stack assembly	JJ	Oil fan drive brush box plug	SS	Turret-to-hull seal
BB	Bilge pump housing outlet jacket	K	Gasket	T	Bilge pump and motor assembly
C	Stack cross brace	KK	Auxiliary engine exhaust outlet seal	TT	Elastic shock absorber cord
CC	Water pump grease	L	Exhaust stack gasket	U	Pump and base gasket
D	Left stack outer brace	LL	Hot spot control shut-off valve	UU	Rangefinder blister cord
DD	Asbestos grease	M	Clamp	V	Bilge pump base
E	Right stack outer brace	MM	Turret race drain plug	VV	Stack release cord assembly
EE	Nonhygroscopic adhesive tape	N	Clamp	WW	Fording cord guide
F	Clamp	NN	Final drive vent line tee assembly	X	Bilge pump hose nipple
FF	Bilge pump housing outlet seal	O	Gun shield cover	XX	Cord guide assembly
G	Exhaust duct support	P	Bilge pump harness	Y	Bilge pump hose
GG	Bilge pump outlet housing	PP	Final drive breather tubing	YY	Fuel tank vent tube
H	Left duct assembly	Q	Fuel tank cap seal	Z	Elbow
HH	Auxiliary engine bellows gasket			ZZ	Left stack outer brace anchor

Figure 3-29. Component Parts of a Typical Deepwater Fording Kit for a Combat Tank

delay may be long enough to permit serious leakage of water into the exhaust system. The problem of inadequate joint sealing in the exhaust system can be alleviated by modifications to the design or by adapter ducts in fording kits (Fig. 3-30). In some model tanks, connections between mufflers and engine are replaced by bellows-type tubes with tight joints. These replacement tubes are included in the deepwater fording kit.

The exhaust end (upper end) of an exhaust stack is normally U-shaped, with the open end facing down to prevent entrance of spray or rain. As an added precaution in maximum depth fording when hydrostatic pressure is greater, the open ends are fitted with restriction plates (Fig. 3-31) having a hole smaller in diameter than the stack proper. The plates provide a restriction to build up a slight back pressure in the exhaust system. This back pressure acts as additional insurance against water entering any joints in the exhaust system. When maximum fording depth is not anticipated, the plates can be removed to elimate the restriction and restore original engine power characteristics.

Since exhaust stacks must be high enough to clear the surface of the water, interference with full 360° gun traverse is unavoidable. Quick-release clamps must be provided at the junction of the exhaust system and the exhaust

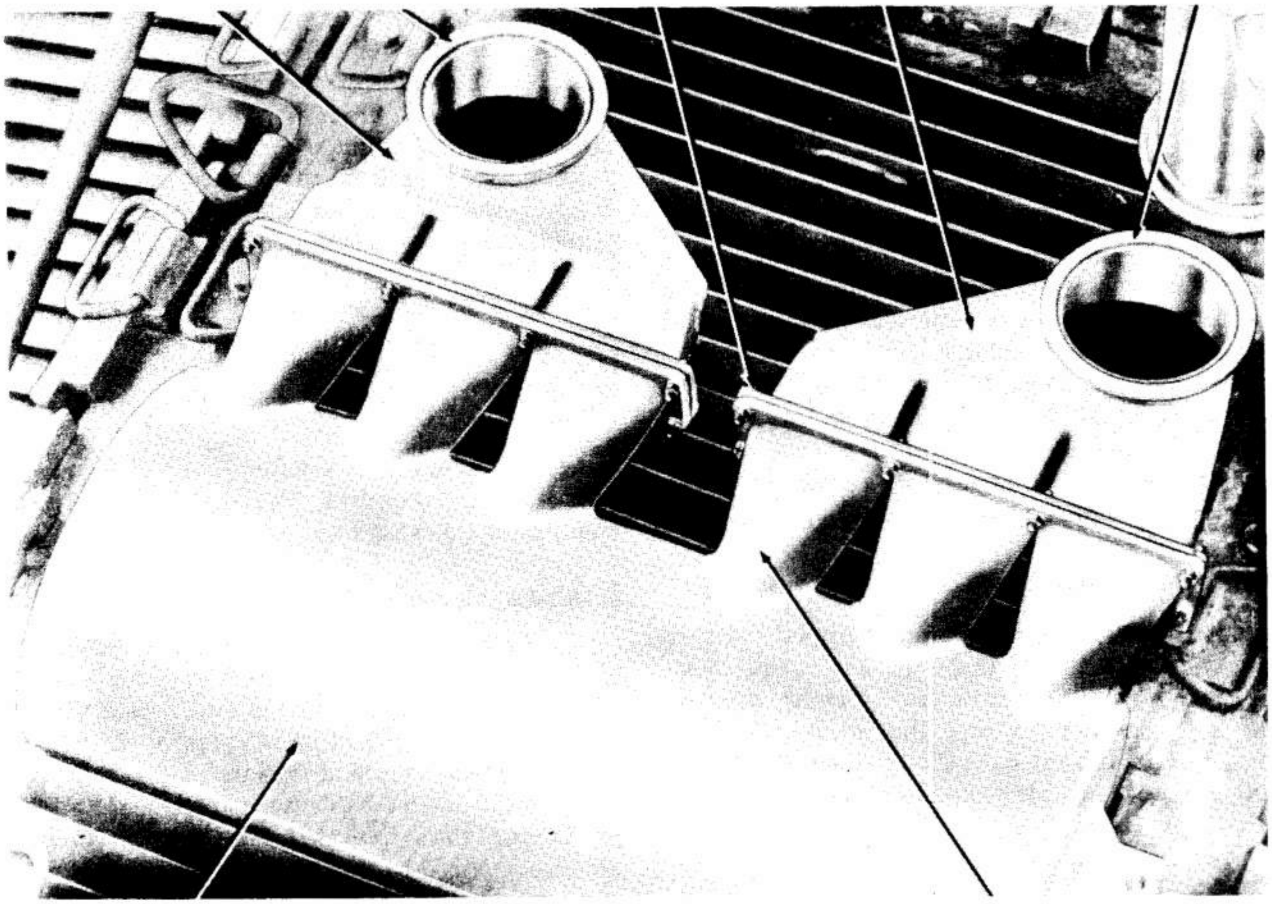


Figure 3-30. Adapter Ducts

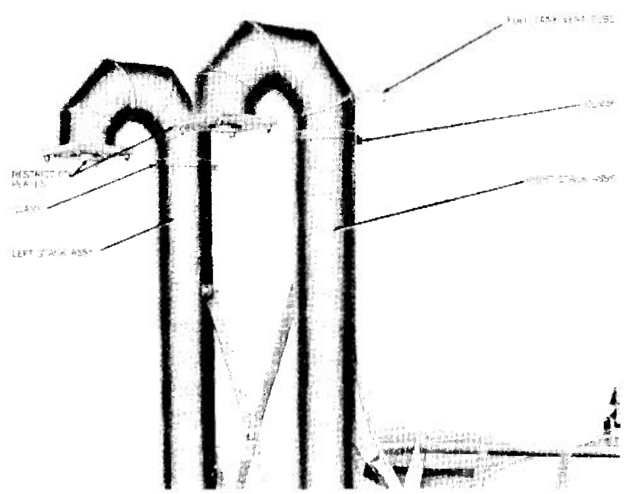


Figure 3-31. Exhaust Stacks With Restriction Plates

stacks to permit jettisoning of the stacks after fording. Current practice is to provide a clamp that can be released by pulling on a rope fastened to the clamp.

Because of their height and light construction, exhaust stacks must be braced. Clevis pin joints are commonly used to assemble the braces to the hull. Ropes fastened to the clevis pins permit the pins to be withdrawn at the time the stack clamps are released. When the clamps are released and the clevis pins withdrawn, the stack and brace assembly falls free of the vehicle and full 360° gun traverse is restored. Rope guides should be provided to prevent the ropes from snagging and becoming inoperative. Simple and effective guides are short lengths of flared tubing assembled to brackets welded to the roof of the hull.

3-36.2.4 Sealing for Deepwater Fording

3-36.2.4.1 Fuel Tanks

Deepwater fording kits must provide for sealing of the fuel tank. Fuel tank filler caps are protected against ballistic attack by hinged armor-steel covers. One method of sealing this type of filler cap is to cement a rubber tube of proper size and shape to the inside of the armor cover. When the cover is closed and latched, the bottom of the rubber tube seats firmly around the filler cap and isolates the fuel tank vent in the cap from water that leaks in around the armor cover (Fig. 3-32). Venting for the fuel tank is provided by a fitting screwed into the armor cover and connected to a hose leading to the open end of the exhaust stack.

3-36.2.4.2 Hot-spot Manifold

Some tank engines are equipped with a hot-spot manifold that utilizes heat from exhaust gases to warm the fuel-air mixture leaving the carburetor. The outlet for the hot-spot manifold is usually located in the engine cooling air exhaust stream. During deepwater fording when the hot-spot manifold ducting is submerged, the value of the hot-spot manifold is questionable because of the cooling effect of the water on the inlet duct. Because its value is doubtful, the hot-spot outlet is normally closed during fording by a seal provided in the fording kit. The flow of exhaust gases through the hot-spot manifold in late model tank engines is regulated by a vacuum control that operates a valve at the exhaust gas outlet. Some deepwater fording kits for engines equipped with this vacuum control include a simple shut-off valve. The shut-off valve is installed between the vacuum control and the hot-spot breather vent tube leading to the intake manifold. When the valve is closed the vacuum control becomes inoperative, thereby closing the hot-spot manifold exhaust gas outlet.

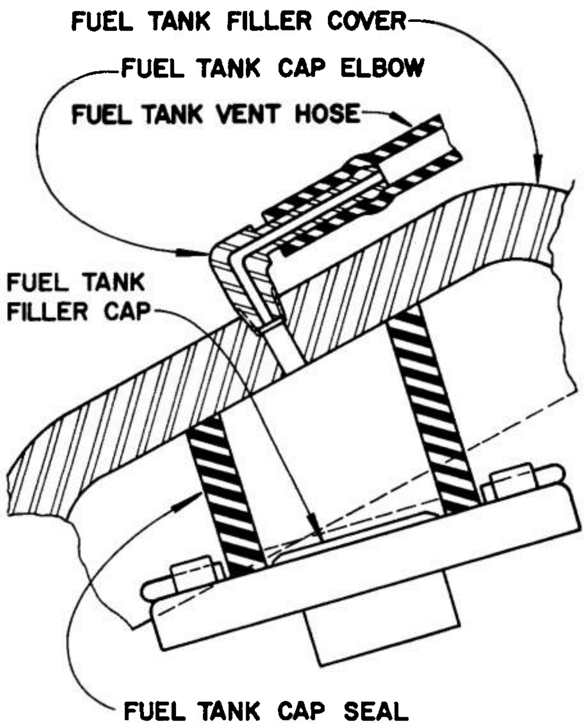


Figure 3-32. Fuel Tank Cap Seal and Vent

3-36.2.4.3 Air Intake System

A watertight air intake system for the engine is important during fording since any appreciable leakage is difficult to detect in time to prevent its affecting engine operation. If leakage occurs in the air stream before it reaches the air cleaner oil bath, water will be deposited in the oil bath and will raise the oil level by displacement. Leakage of any considerable size, or extending over an appreciable duration of submersion, can raise the oil level enough to seriously restrict the air supply.

One of the problems to be solved in the waterproofing of the air intake system is the location of the air cleaners. From a fording standpoint, the best location is in the fighting compartment where a minimum number of joints are exposed to water pressure. Of course, other design factors are involved in locating the air cleaners, but fordability should be an important consideration.

Associated with the location of the air cleaners is the matter of routing the air to and from the air cleaners. If provision for drawing air from either the crew compartment or engine compartment is to be maintained, the means of switching from one source to the other must be simple, convenient, and capable of positive sealing.

3-36.2.4.4 Bilge Pumps

Formerly, nearly all combat tanks were equipped with two bilge pumps, one on the

floor of the engine compartment and one on the floor of the crew compartment. Both bilge pumps are omitted in current tanks. This omission is an economy measure because bilge pumps are considered unessential except in deepwater fording operations. Consequently, a bilge pump must be included in every deepwater fording kit for installation before deepwater fording. Design of a bilge pump installation for a deepwater fording kit is concerned primarily with the location and sealing of the outlet hose. To avoid installing a new outlet for the bilge pump hose in the roof of the tank hull, one of the driver's periscope openings is utilized in some installations. The periscope mount is removed and replaced with an assembly to seal the opening and provide an outlet for the bilge pump hose fitting. More recently, designers have taken cognizance of the bilge pump outlet problem during deepwater fording operations by providing an opening in the tank hull roof to one side of the driver's hatch. An elbow can easily be mounted over this prepared hole which is covered with a plate until the elbow is installed. A one-way valve must be provided at the outlet to prevent the entrance of water should the bilge pump become inoperative. A simple way to accomplish this one-way action is to use a rubber valve of the duckbill type. A rubber duckbill valve provides simple one-way action with external water pressure furnishing the sealing pressure.

3-36.2.4.5 Auxiliary Engine Exhaust

On some recent model tanks, the muffler for the auxiliary engine exhaust is mounted on top of an engine access grille. Raising the grille separates the face-to-face connection between the bellows-type auxiliary engine exhaust outlet and the muffler inlet. In normal operation, the seal between the exhaust outlet and the muffler is adequate. However, during deepwater fording, the seal is susceptible to water leakage. More efficient sealing is effected by placing a thick gasket between the exhaust outlet and the muffler. This thick gasket causes the exhaust outlet bellows to exert a greater sealing pressure. Although the auxiliary engine is not operated during fording, operation immediately after fording might be necessary. To seal the muffler outlet and still permit immediate operation of the engine after fording without requiring removal of the seal, a rubber duckbill valve is provided. When flexible tubing is used to connect the exhaust outlet and the muffler, the flexible tubing is disconnected and a duckbill valve installed on the exhaust outlet.

3-36.2.4.6 Turrets

Tank turrets present several sealing difficulties. The opening between turret and hull is one major problem. In some tanks, the opening between turret and the hull can be sealed with a triangular rubber strip. The sealing strip is coated with grease, forced into the opening between the turret and hull, and held in place with a strap or a band of elastic shock absorber cord. A turret-to-hull seal that can be installed at the time of manufacture and remain in place at all times has been developed (par. 3-8.6). Holes are provided in some tank hulls just below the turret race to permit draining of water that might enter the turret race during rainy weather operation. These drain holes must be plugged before deep-fording is attempted. Usually, simple rubber plugs suffice to seal these drain holes.

Openings in and around the gun shield present a particularly difficult sealing problem. Since the tank gun must be able to fire at all times, gun shield sealing must be flexible enough to permit elevation and depression of the primary armament. At the present time, the most effective gun shield seal consists of a waterproof canvas cover (Fig. 3-33). The edge of the cover is clamped tightly to the front face of the turret, around the gun shield and the short tubular section of the cover is clamped securely around the gun tube. The tubular section of the cover must be long enough to permit recoil of the gun. Individual pockets are provided in the cover to accommodate the sighting telescope, the coaxial machine gun muzzle, and the gun shield lifting eyes.

The waterproof canvas-type gun shield cover has several serious drawbacks. Among the more important are its vulnerability to ballistic attack and the loss of sealing once the coaxial machine gun is fired. Due to the short life of these covers in the field, each fording kit provides a complete cover assembly for installation in the field as needed. Development of permanent, built-in sealing would be a major improvement.

Sealing of the gun muzzle and rangefinder blisters can be readily accomplished by the use of nonhygroscopic adhesive tape. The gun

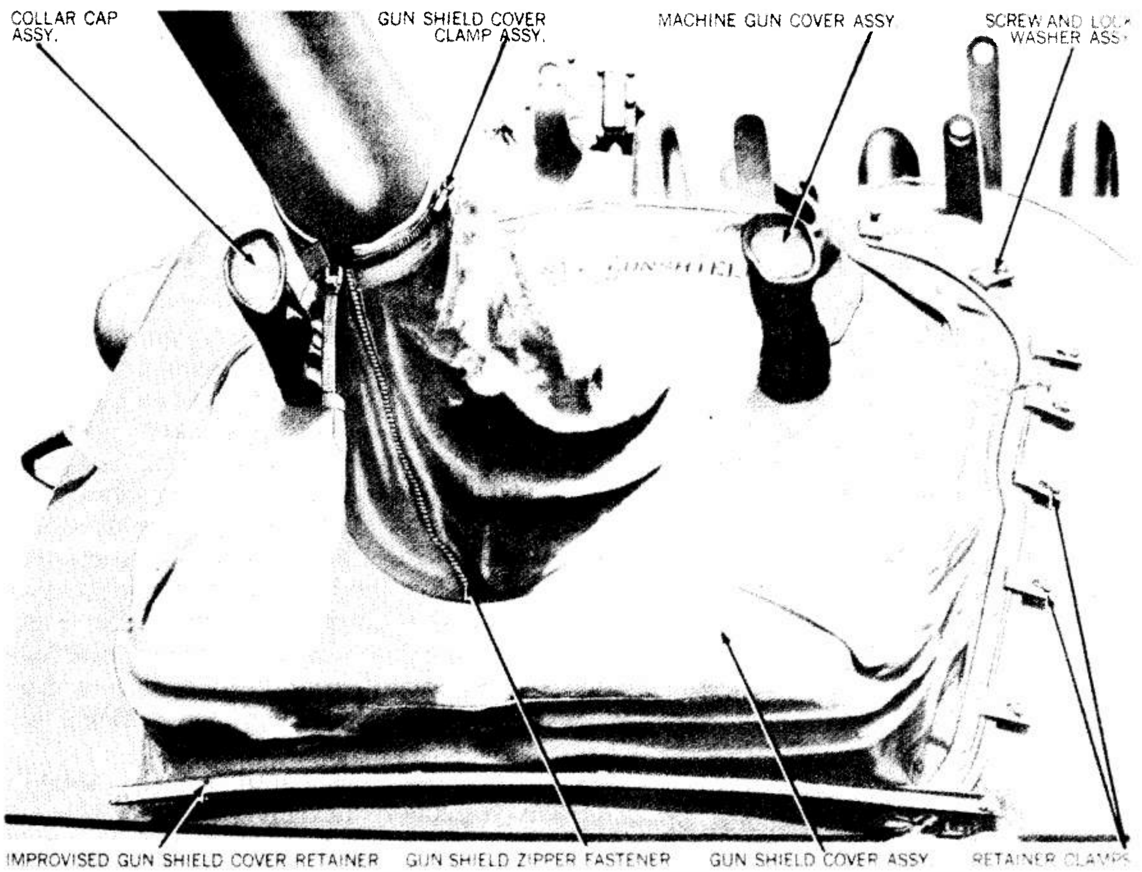


Figure 3-33. Gun Shield Cover Assembly

muzzle sealing can be perforated by firing an armor-piercing round when the gun must be used. Rapid removal of the rangefinder blister sealing tape can be accomplished by taping over the knotted end of a rope placed over the blister opening. Pulling on the rope instantly strips off the sealing tape.

Fire extinguisher horns must be sealed to prevent water damage to automatic valves in the fire extinguisher system. Snap-on rubber caps have proven to be effective seals without rendering the extinguisher system inoperative.

3-36.2.4.7 Miscellaneous Openings

Miscellaneous minor openings that must be sealed prior to fording are readily sealed with either asbestos grease or nonhygroscopic adhesive tape. However, because of their number and unobtrusiveness, minor openings can easily be overlooked when preparing the tank for deepwater fording. Therefore, designers should attempt to provide inherent sealing in as many components as possible to avoid the necessity of extensive taping and greasing before fording.

3-36.3 FLOTATION DEVICES

During landing operations, troops and their equipment are particularly vulnerable to enemy fire until they have arrived on the beach and set up their weapons for defense. To supplement naval gunfire and air support, it is desirable to use the firepower of tanks participating in the landing operation. Firing the weapons of a tank equipped with a deepwater fording kit is possible only if the water level is below the gun muzzle. Flotation devices (Fig. 3-34) plus deepwater fording kits are necessary to float a tank high enough in the water to permit firing the tank armament. Flotation devices in conjunction with deepwater fording kits are also used to enable tanks to negotiate deep rivers and lakes when better facilities are not available.

Experience gained from amphibious operations in World War II and subsequent tests

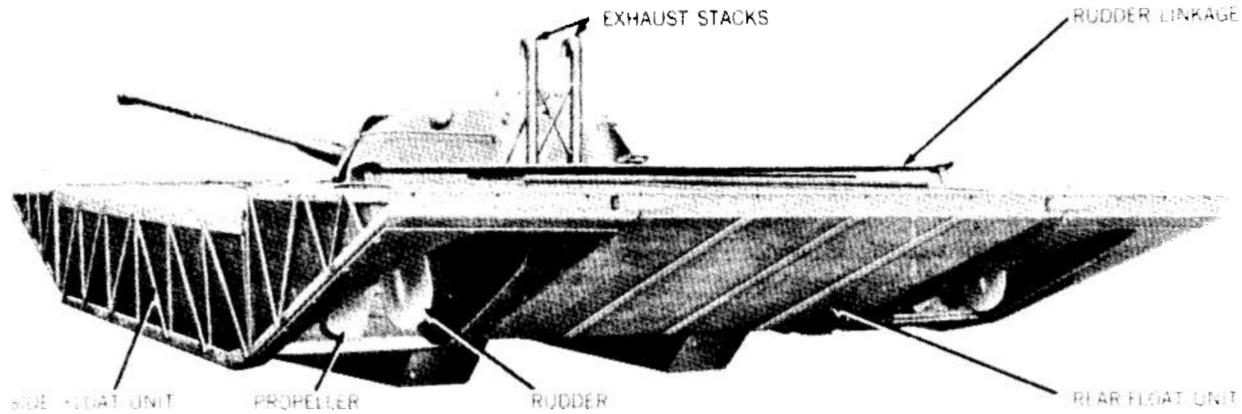


Figure 3-34. Flotation Device

have established the following military requirements for a successful flotation device:

a. Weight to be such that component parts can be handled for attachment by a double tank crew, aided by the equipment normally assigned to maintenance shops of tank battalions or similar installations, with a minimum of special tools and in a minimum practicable time.

b. Length such that the tank can negotiate a 40-percent (22°) slope either above or below the surface of the water, with the flotation device attached and operable.

c. Overall width when attached to the tank to be the minimum practicable.

d. Height such that all tank armament can be fired through a field of fire of 360° when the flotation device only is attached. The exhaust stacks of the deepwater fording kit may limit the field of fire.

e. When disassembled and inoperable, the flotation device must be capable of being transported in quantity in standard wheeled or tracked cargo carriers, or by air.

f. The flotation device to consist of a metal frame (steel, titanium, or aluminum) assembly in which an approved plastic material, or an approved substantially equal material, can be used to supply the required buoyancy to float the tank.

g. The plastic material to have a maximum density of 3 lb/ft^3 and a minimum compressive strength of 30 psi. The plastic material must be resilient and shock-absorbing, waterproof, fire-resistant, chemical-resistant, vermin- and fungus-proof, and have a maximum heat shrinkage of one percent.

h. The metal frame structure supporting the flotation units to be made detachable from the tank by use of explosive pins, controlled and discharged by the tank operator.

i. Flotation blocks to be securely locked into four units by the metal structure, one each for the front, rear, and sides of the tank. Each unit to be discharged at will by the tank operator according to a predetermined sequence.

j. Flotation units must provide ground clearance, approximately two inches above that of the bottom of the hull of the tank, exclusive of bumpers.

k. Provisions shall be made for protection of the bottom and sides of flotation units to prevent or lessen damage to flotation blocks from the road or when crossing obstructions.

l. Angle of approach of departure on front and rear floats to be not less than 20°.

m. Greatest degree of interchangeability of various parts of flotation units must be provided and maintained using standard and uniform-size float blocks and metal structure.

n. Propulsion in the water to be by propellers on both sides of the tank powered by power take-offs mounted in the drive sprockets, with means of disconnecting the propeller drive during operation of the tank on land and water.

o. The flotation device must maintain flotation and proper trim of the tank in streams with current speeds of 11 fps (7.5 mph).

p. The speed of the flotation device to be not less than 6 mph in smooth still water.

q. The flotation device must be sufficiently durable to withstand normal handling and cross-country travel while attached to the tank

or in the course of being transported by truck, train, or otherwise.

r. The flotation device shall be designed to maintain flotation with the tank hull completely filled with water.

s. The flotation device must be capable of maintaining flotation of the tank after multiple punctures by cal .50 or cal .30 bullets.

t. Jettisoning must be accomplished from inside the tank.

u. Steering must be accomplished from inside the tank.

Although the foregoing requirements have been established for tanks, many of them apply to flotation kits for other tracked and wheeled vehicles as well.

3-36.4 WINTERIZATION KITS

3.36.4.1 General

Experience in World War II and Korea emphasized the need for military vehicles capable of operating in any geographical area during any season of the year. In recognition of this need, AR 70-38 states that all developed materiel should be capable of acceptable performance throughout an ambient air temperature range of 110° to -25°F without the aid of any accessory or device that is not a part of the standard vehicle, and down to -70°F when modifield or augmented by suitable materials or devices from a winterization kit developed for this purpose.

Extreme cold weather operation exposes vehicle components and equipment to numerous hazards. Conventional lubricants and fluids thicken or congeal and, except for a few special types, must be heated or diluted to permit proper operation. Electrical systems fail, and tactical radio communications may be completely disrupted. Frost on high-tension wires may cause shorts if insulation is poor or deteriorated. Battery cases become brittle and battery charge runs low because of increased use of electrical accessories.

Increased strain is placed on batteries when starting engines at low temperatures. As temperature decreases, chemical reaction in a battery is retarded and the battery loses its efficiency both in output and ability to accept a charge. The state of charge of a battery plays an important part in preventing freezing of the electrolyte. A fully charged battery with specific gravity of 1.275 to 1.300 will not freeze at -80°F, but a discharged battery will freeze at 10°F.

At extremely low temperatures, standard fuels do not vaporize and combine properly with air to form a combustible mixture for starting. Winter-grade gasoline overcomes many faults, but not the problem of icing and freezing of fuel lines and filters. This trouble is caused by condensation, by snow entering the fuel tanks or containers, or by the water that always seems to be present in containers in which fuel is supplied. Straining fuel through chamois skins, keeping tank caps firmly in place, and even keeping fuel tanks full do not prevent some water from entering the systems. Alcohol added to the fuel combines with the water and allows it to pass into the engine and be burned.

Condensation is a serious hazard not only to fuels but to crankcases, transmissions, and fire control and sighting instruments as well. In transmissions and crankcases, condensation may cause icing and increase wear of gears, bearings, and other parts, and may cause control levers to freeze in position when vehicles are parked.

Fire control and sighting instruments, especially periscopes, are susceptible to condensation as a result of their physical placement in the vehicle. In cold weather operations, one end of the instrument is exposed to cold outside air while the other end is exposed to warm, moist air inside the vehicle. The warm, moist air enters the optical tube of the instrument and condenses on the colder surfaces of the lenses. Fogging results and the instrument becomes useless. Protective coatings have been tested, but such coatings tend to affect the optical properties of the instruments. A new type electric conductive coating that heats the lens has also been tested, but results have not yet been substantiated. Condensation on fire control and sighting instruments also may be caused by breath of the operator or by transferring the instruments from cold to warm air. Moisture thus collected may cloud optics and start internal rusting. If exposed to cold air, the moisture may freeze and set up strains in the working mechanisms. All armament and components must be kept dry and properly (usually "lightly") lubricated to function in extreme cold.

Other difficulties resulting from exposure to extreme cold are materials become brittle, routine maintenance becomes difficult and requires up to five times as long to accomplish as the same task under normal conditions, tracks freeze to surfaces, frozen streams and deep snow drifts become difficult to cross, and general physical discomfort lowers the efficiencies of crew members and maintenance men.

Winterization kits are sets of equipment designed to extend the cold weather capabilities of specific vehicles or equipment by either protecting temperature sensitive components from the effects of the cold or by providing heat to offset the cold. The compositions of the kits vary greatly and depend upon the types and characteristics of the vehicles or equipment for which the kits are intended. Thus, winterization kits for open-cabbed vehicles may include side curtains or panels to enclose the cab or, for cabless vehicles, they may include completely insulated cabs for the protection of the operator or crew along with varied other equipment. Practically all winterization kits include some device for applying heat to the vehicle to insure starting of the power plant and warming vital components, suitable ductwork and tubing to distribute the heat more effectively, an assortment of hardware and miscellaneous fittings to facilitate field installation of the kit components, various canvas covers with which to close grilles and other openings from which heat might escape, and may even include a tent to completely enclose the vehicle during warmup. The principal functions of power plant heaters are:

a. To warm the engine oil above its pour point so as to reduce the cranking effort and provide immediate lubrication to the engine

b. To warm the engine block and thereby reduce cranking friction

c. To provide warm carburetor or combustion air in order to volatilize the engine fuel

d. To warm the battery above 0°F in order simultaneously to secure minimum cranking speed and ignition voltage

It is important to analyze the components of the winterization equipment and weigh their requirements for field installation. Whenever an excessive number of man hours are involved by the field installation of any component, that component should be included in the basic design and produced as an integral part of the basic vehicle together with all necessary mounting blocks, holes and clearance cutouts, electrical outlets, and fuel-line connections. The remaining items may constitute a kit to be installed in the field as required.

3-36.4.2 Starting Aids

3-36.4.2.1 Classification of Starting Aids

The principles of applying heat to motor vehicles to insure starts are well developed. Two general methods of heating power plants for starting are used; these are *standby heat* and *quick heat*.

The *standby heat* method uses a comparatively small heater that operates continuously while the vehicle is idle. It must produce sufficient heat to compensate for losses and keep the power plant from becoming cold. For standby heating of vehicles having engine displacements of 100 to 300 $in.^3$, 20,000 Btu/hr properly distributed will maintain satisfactory temperatures at all desired points. If the engine has already become cold, sufficient time must be allowed to bring it to starting temperature. This condition occurs when materiel is shipped into a region where heated buildings are not available. When standby heat is used, the vehicle is always warm and ready to start. Heat is usually supplied to liquid-cooled power plants through a thermo-syphon system. In this way, the use of pumps and fans that drain batteries is avoided. Heat can be supplied to batteries by hot water coils. This method avoids the danger of over-heating. Batteries can be kept warm more easily than they can be reheated when once allowed to get cold. Since space is usually at a premium, the small size of the standby heater is a distinct advantage.

The *quick heat* method is very well suited to warming up the current family of air-cooled tank engines. It permits these engines to be started in less than one hour after having been thoroughly cold-soaked at −65°F. This performance requires a heat input of 30,000 to 40,000 Btu/hr; although the heating of large equipment may require as much as 100,000 Btu/hr.

Several design problems are presented by quick heat. One of these is the prevention of damage to electrical equipment, another is delay

in warming the battery. Conventional rubber cased batteries cannot be heated at a rate faster than about 1 °F per minute. Supplying heat at a rate faster than this overheats the battery case and may damage the battery. Approximately 45 minutes are required to warm a battery to −20° F from −65°F at 1° per minute. In the field, however, the inside of batteries seldom, if ever, actually reach −65°F, as it takes a long time to soak down to ambient temperature. Field starts, therefore, may often be made in 30 minutes or less. Special batteries do exist which provide for rapid heating. Quick heat eliminates the need to heat equipment that is out of service, and heaters are used only when the operator is present. Life of the quick heater is greater and maintenance requirements are less than for other types requiring constant operation. If no electric current is available, an auxiliary cold starting aid (slave kit) must be used.

Both standby and quick heaters have advantages and both are in use today. The present trend is toward a combination of the two. Such a combination heater should be powerful enough to heat a thoroughly cold-soaked power plant from −65°F to a starting temperature in 45 minutes to one hour. The heater should be thermostatically controlled (on-off or high-low) so it can be used for standby heat if desired with no danger of overheating coolant, battery, or other sensitive components.

3-36.4.2.2 Combined Power Plant and Personnel Heating

Power plant heating systems may be combined with personnel heating systems and made to perform both functions. Combined power plant and personnel heating offers several advantages. For example, one heater replaces two with possible savings in cost, parts to be serviced, space occupied, maintenance time, and in the training of operators. Finally, since power plant heating and personnel heating are seldom required at the same time, the combined heater does not need to be much larger than either individual heater.

3-36.4.3 Types of Winterization Equipment

Under-chassis heating was one of the early systems used to facilitate starting. The method was simply to place a space heater without electric power near or under the vehicle and cover vehicle and heater with a tent or tarpaulin. The difficulty of handling and storing stiff frozen canvas covers, keeping burners going, and of preventing fires made under-chassis heating impracticable for combat conditions. To remedy this situation, small, compact heaters that burn readily available gasoline and can start and cycle automatically under control of thermostats have been developed. These heaters consume a minimum of fuel and have proven sufficiently practical to be standardized.

Electric heating elements applied at points where the temperature must be raised is one of the simplest methods of getting heat to the right places. Because of power requirements, however, electric heating is limited mainly to vehicles stationed at permanent posts. Standby heating for most vehicles requires a heat input of 15,000 to 25,000 Btu/hr. This would require about 4 to 7 kw input to the heater of each vehicle; and for quick heating, this power requirement would double. Engine-generator sets can be incorporated into the vehicles to supply this power. However, this system is costly and has the additional disadvantage of requiring heating in cold weather before it can be started.

Portable heaters containing their own batteries or operating from the vehicle's batteries have been considered as means of warming up cold-soaked equipment. Heated air can be delivered through flexible ducts to the areas needing warming, or heat ducts can be an integral part of the vehicle and the portable heater can be connected temporarily to them. Most portable heating systems employ an auxiliary battery. Hand cranked portable heaters have been tried to eliminate the need of electrical power. The exertion of cranking at ambient temperatures of −40°F and colder, however, so increase the operator's rate of breathing that he is in danger of freezing his lungs.

The *Cold Starting Kit (Slave Kit)*, M40, provided with 6-v, 12-v, and 24-v battery boosting circuits that can be connected to the electrical systems of vehicles or to other equipment to facilitate starting. The kit also includes a gasoline engine-driven generator for battery charging and a gasoline burner for supplying a large volume of heat for use as a quick starting aid. The high capacity of this system makes it possible to start average sized vehicles in as little as five minutes and even large tanks can be put into operation in a very short

time. When the slave kit itself is cold, its batteries and engine generator can be warmed up with the self-contained heater. The complete system can be transported on a light cargo truck or dragged for short distances on skids.

Contaminated air-type heaters are also used to warm cold-soaked power plants. These are equipped with burners that have high excess air (low CO) to produce a large volume of comparatively low-temperature exhaust gases. These exhaust gases are ducted to critical points within the engine compartment. Usually, the air is discharged beneath the engine and allowed to rise up through the cylinder fins and the air cooling system of the engine. Every precaution must be taken when this system is used to prevent carbon monoxide from entering the crew compartment. Batteries are enclosed in an insulated box through which the contaminated air is circulated to heat the batteries rapidly. The system requires thermostatic valves to cut off the heat before the batteries are damaged.

Standby contaminated air heating of air-cooled engines requires no external fans or pumps and, thus, has a very low current drain. quick heat, contaminated air heating is rapid and efficient. Its main disadvantages are condensation and its resultant corrosion, the danger to personnel of breathing the exhaust gases, and the necessity for installing stainless steel ducts.

Radiation heating is a modification of the contaminated air system. In this system, the heater exhaust gases are carried through ducts to heat exchangers located adjacent to the points to be heated. Power plants are thereby protected from direct contact with the contaminated gases. A combination of radiant and direct heating with exhaust gases could be designed to obtain high heating efficiency with protection for personnel and all delicate parts of the equipment. Such a system would be more expensive and more difficult to install.

Recirculating fresh air heaters can also be used effectively to warm air-cooled engines. Such heaters are equipped with large heat exchangers and high capacity blowers. Clean air is directed through the heat exchanger and the hot air is ducted to the areas to be heated. Since only clean air comes in contact with the equipment being warmed, the corrosion problem is greatly reduced. Batteries are heated by circulating the heated air through an insulated jacket, or box, surrounding them.

Space limitations require that the blower be as small as practical; therefore, in order to achieve high capacity, it must operate at a high speed. This tends to increase the noise problem and the current drain—both of which should be kept at a minimum. Distribution of the heat to all the necessary areas also presents design problems. Some designers have proposed blowing some of the heated air directly into engine crankcases and gearboxes. This scheme requires that dirt and moisture be removed from the air first and that some means of preventing overheating the oil in crankcases be incorporated into the system. Also required is a means of preventing loss of engine oil through the heating air ducts when the engine is in operation.

3-36.4.4 Personnel Heating

Under field conditions in arctic regions, military personnel dress in arctic clothing that allows them to live and fight in outside temperatures. In even a moderately heated area, they must remove their arctic clothing or be drenched with perspiration. If forced to return to the outside in sweaty clothing, they may suffer frostbite or freezing. The personnel heater, therefore, should only make up the difference in body heat between moving and sitting still, or allow for a partial removal or loosening of outer clothing. Under present requirements, tank personnel heaters must be capable of heating the fighting compartment to 20°F and the driver's compartment to 10° F from -65°F, after one hour of operation.

Tanks are currently equipped with gasoline-fired forced air heaters rated at 20,000, 30,000 or 60,000 Btu/hr. The heater and its control panel are usually installed in the driver's compartment. Asbestos-covered ducts direct the flow of heated air to where it is needed and dampers regulate the air flow.

3-36.5 DESERTIZING EQUIPMENT

Desertizing equipment is considered to be those appliances and modifications that are necessary to equipment to assure its satisfactory operation in the desert and in other locations where extremely high temperatures (up to 125°F) are experienced. The main difficulties encountered in high-temperature operation of

vehicles are the formation of a vapor lock in the fuel system; overheating of the engine and electronic equipment; and, in desert areas, the induction of dust into the engine and crew compartment. Some of the items that might be included in a desertizing kit are:

a. Electric fuel pump to replace the standard fuel pump. The electric fuel pump is more effective in preventing vapor lock.

b. Radiator surge tank for liquid-cooled engines to contain the overflow of the cooling liquid as it expands during hot weather operations.

c. Larger diameter cooling fans for air-cooled engines.

d. Cooling equipment for heat-sensitive instruments.

e. High-capacity air filters for both the engine and crew compartments.

3-37 SPADES, BOOMS, WINCHES, AND BULLDOZERS

3-37.1 SPADES

A spade, as the term is used here, is a component of a self-propelled gun or howitzer. Its function is to transmit the gun recoil reaction forces, experienced by the vehicle every time the weapon is fired, into the ground. This makes the vehicle a fairly stable firing platform, even in relatively soft ground, and prevents the recoil reaction load from being transmitted into the vehicle drive train. A further function of the spade is to simplify and speed the emplacement of the weapon. Figs. 1-24 and 1-30 show two types of self-propelled artillery equipped with spades. These guns are emplaced simply by lowering the spade to the ground and then backing up the vehicle until a sufficient spade resistance is obtained. The teeth of the spade aid in digging in the blade, particularly in hard or frozen soil. When the weapon is to be moved, the vehicle is simply driven forward until the spade is free of the soil, the spade is raised, and the vehicle is ready to travel.

Spades similar to those used to stabilize self-propelled artillery are also provided on heavy recovery vehicles. Their function here is to provide sufficient reactive force to resist heavy horizontal winching forces or to help stabilize the vehicle against tipping forces when making heavy lifts with the boom. The recovery vehicle shown in Fig. 1-26 is equipped with two hydraulically operated spades—one fore and one aft.

3-37.2 BOOMS

Booms are structural devices mounted upon certain vehicles to give them the capability of lifting heavy loads vertically and to position these loads laterally or longitudinally with respect to the vehicle axis. They are found on such vehicles as wreckers (Fig. 1-12), recovery vehicles (Fig. 1-26) and certain engineer vehicles—cranes, power shovels, draglines, etc. The general configuration of the boom, its length, and the load capacity required are dependent upon the mission of the vehicle and the purpose of the boom. Booms on engineer vehicles are used to perform medium duty tasks, such as lifting and moving bridge sections, pontoons, heavy timbers, miscellaneous cargo, and even lightweight vehicles; while the boom of the heavy recovery vehicle shown in Fig. 1-26 is capable of lifting 30 tons. In addition to the factors normally considered in any design—such as the forces involved, structural strength, material selection, and the design configuration—the boom designer should also consider the following:

a. The restrictions imposed by the vehicle's suspension system—e.g., the load limit of the spring system; the stability of the boom base when loading, unloading, and swinging the boom; the need for, or availability of, suspension lockout provisions on the vehicle; limitations on the reach of the boom due to the elastic support; and the need for outrigger devices to prevent tipping the vehicle.

b. Wherever possible, the boom should be designed with the capacity to swing the load laterally as far as possible by rotating about a vertical axis. A full 360° swing capability is most desirable but is not always practicable. Mounting the boom in the turret of a tank-type vehicle, or providing it with a rotatable mounting platform, simplifies the swing requirements.

c. When designing booms for amphibious vehicles, the effect of the boom weight and location of its center of gravity upon the vehicle trim in the water and its freeboard should be carefully evaluated. This is particularly important if the booms will be operated while the vehicle is afloat.

3-37.3 WINCHES

The primary purpose of winches on military vehicles is to improve their mobility by providing a means by which they can pull themselves, or other vehicles, out of difficulty when they become immobilized by adverse terrain. For this purpose, they are usually mounted at the front of the vehicle—immediately behind, or on top of, the front bumper (Figs. 1-9 and 1-27). This location has several advantages over a location at the rear—the operator has better visual control over winching operations; winching operations will not be hampered by trailers or other towed loads that are commonly connected to the rear of military vehicles; the winch is generally more accessible at the front of the vehicles; and the winch is less susceptible to damage from rocks, logs, stumps, and other rough-terrain hazards. In addition to the primary purpose stated, expediency finds many other uses for vehicle-mounted winches. They have been used to snake logs out of inaccessible places; to pull artillery into difficult positions; to operate improvised hoists, derricks, and high-lines; and countless other applications that the ingenuity of troops in the field can devise.

Winches are also used on special-purpose vehicles—such as wreckers, recovery vehicles, motorized cranes, power shovels, pile drivers, etc.—to operate booms and hoisting equipment. For these applications, they are mounted at the locations most advantageous to the performance of their specific function. This generally places them at various locations behind the vehicle cab (Figs. 1-13 and 1-14). Table 3-13 lists the drum-type winches currently being used on military vehicles. Most winches on military vehicles are driven mechanically through a power takeoff attached to the main drive train; although hydraulic drives are also used. Fig. 3-35 shows a typical power takeoff and winch drive.

3-37.4 BULLDOZERS

Bulldozer blades are often attached to tanks, tractors, and even trucks to give these vehicles the capability of clearing land of vegetation or debris; to assist in the preparation of weapon emplacements, breastworks, access roads, or fire lanes; or to perform any number of related tasks. These attachments consist of a large moldboard mounted across the front of the vehicle, two push beams, four tilt arms, two carrying hooks, two double-acting hydraulic cylinders, and suitable mounting brackets. Guards must be provided to protect all vulnerable lines and components that are located outside the vehicle hull. The moldboard is supported by a quadrilateral linkage consisting of the two push beams, which are connected near the bottom of the moldboard, one on each side, and two inner and two outer tilt arms connected to the moldboard. The controls for the bulldozer assembly should be located conveniently to the vehicle driver to enable him to drive the vehicle and control the blade simultaneously. The following factors should be considered when designing bulldozer attachments for vehicles:

a. The weight of the bulldozing blade and its accessories must be kept to a minimum. When the attachment is in place, the suspension loading of the vehicle is changed and the original weight distribution and balance of the vehicle may be upset. For this reason, bulldozer attachments do not lend themselves readily to light vehicles. Aluminum blades have been tested, but the results of these tests were inconclusive as to the suitability of these lightweight units.

b. Bulldozer blades are usually hydraulically operated, and provisions must be made for a source of hydraulic power. The hydraulic components must be designed to minimize leakage and vulnerability and be sufficiently durable to permit cross-country operations with the blade in the travel position.

c. An emergency blade-lifting device or mechanism must be incorporated into the design to enable the blade to be raised in the event of a hydraulic failure.

d. The bulldozing attachments are normally designed to be mounted on the towing lugs of the vehicle. This arrangement is not entirely satisfactory. Special mounting lugs may improve the overall performance of the bulldozing equipment.

e. The bulldozing blade should not obstruct the visibility of the vehicle operator when it is in the raised position.

3-38 FUEL TANKS

Fuel tanks should be as large as possible to secure the maximum cruising range for the

TABLE 3-13 DRUM-TYPE WINCHES USED ON MILITARY VEHICLES

Rated Capacity, lb	*Cable Dia, in.*	*Cable Length, ft*	*Vehicles on Which Used*	*Approx. Net Wt* of Vehicles, lb*
5,000	3/8	200	Amphibious Cargo Carrier, M76 and ¼-ton trucks	2,200 to 8,800
7,500	7/16	300	¾-ton trucks	6,000 to 7,000
10,000	1/2	300	2½-ton trucks, truck tractors, wreckers	11,000 to 24,000
20,000	5/8	300	5-ton trucks, truck tractors, wreckers	22,000 to 40,000
35,000	3/8	300	12-ton Truck Tractor, M26A1	48,900
45,000	3/4	300	10-ton trucks	30,000
50,000	7/8	300	Hoisting winch of M88 Recovery Vehicle	—
60,000	7/8	300	12-ton Truck Tractor, M26A1	48,900
90,000	1-1/4	200	Main winches on M51 and M88 Recovery Vehicles	112,000 and 113,000

*Weight of empty vehicle

vehicle. A minimum cruising range is often specified in the procurement documents. The fuel tanks may be positioned in any convenient location; however, fuel tanks in amphibious vehicles should be located so as to minimize the shift in vehicle trim in the water caused by fuel consumption. Factors such as vulnerability, effective space utilization, and convenience in servicing should also be considered when selecting tank locations and fuel tank materials. Since all military automotive engines are equipped with fuel pumps, the elevation of the fuel tank may be less than that the carburetor. Typical fuel tank locations are under the floor, in the walls, and in the ceiling of the hull. Wall and ceiling locations have the disadvantages of increased vulnerability to ballistic penetrations; although they do impart a measure of protection against nuclear radiation. Dual fuel tanks are sometimes used to decrease tank vulnerability and increase fuel system reliability.

A prime consideration in fuel tank construction is that it be resistant to corrosion. Some popular materials used for this purpose are Fiberglas, rubber, and plastics; although the most common fuel tank materials are sheet steel, with a corrosion-resistant plating, and aluminun. Tanks made of nonrigid, materials should be properly supported by rigid structures. Integral fuel tanks, while attractive from a space utilization point of view, should be avoided because of the repair and maintenance problems associated with their use. In order to minimize damage from shocks, fuel tanks are usually shock mounted.

The inlet for filling the tank should allow a minimum filling rate of 50 gpm and should be vented to admit air to the tank as the fuel is used. A fuel filter should be incorporated into the tank inlet to remove fuel contaminants. The configuration of the filling inlet should assure that fuel spilled during filling will not flow into the engine compartment.

The configuration of the tank outlet should be such as will prevent sediment collected on the tank bottom from being drawn into the fuel line. A minimum depth of ½-in. should be allowed for a sediment trap. A drain plug should be provided for draining and cleaning the tank and should be located so that the fuel will be drained overboard rather than to the inside of the vehicle. Electrical fuel pumps are in common usage on military vehicles because of their greater reliability and their greater effectiveness in preventing vapor lock when highly volatile fuels are used. Fuel lines should be run as directly as possible, avoiding numerous and sharp bends. They should be well supported and run through protective grommets at bulkheads to prevent chafing due to vibrations.

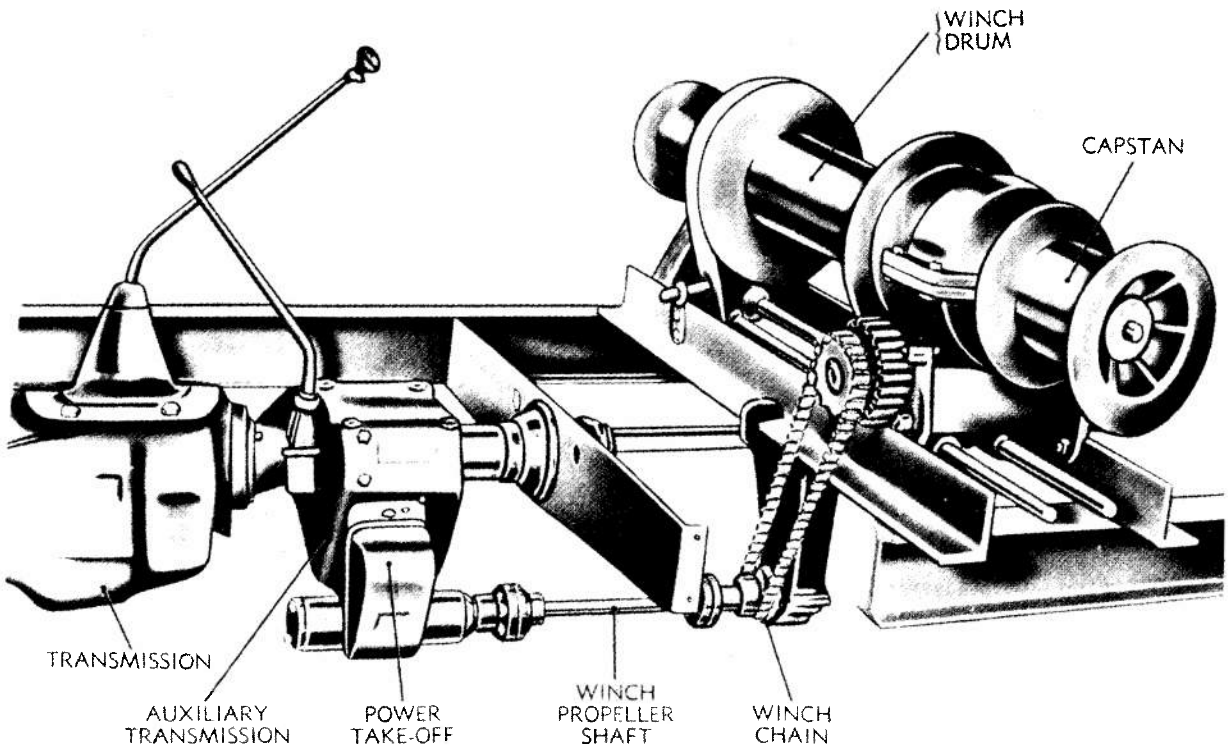


Figure 3-35. Auxiliary Transmission, Power Take-off, and Winch Assembly

Figs. 3-36 and 3-37 illustrate what must sometimes be done to get maximum fuel tank capacity with limited available space. The figures show the left- and right-hand fuel tanks of the M60 Combat Tank, respectively. These are located in the engine compartment, one on either side of the power plant. Their rather elaborate shapes are the outcome of trying to utilize all available space for fuel and result in a combined capacity of 385 gal for the two tanks. They are fabricated of riveted and welded aluminum plates and extrusions and are provided with internal baffles to reduce sloshing of fuel. Each tank is shock mounted at three points.

3-39 AMMUNITION STOWAGE

Most combat vehicles require the onboard stowage of some ammunition. In the case of tanks, self-propelled mortars, and smaller caliber self-propelled weapons, all of the ammunition needed for a typical mission is carried on board. On the other hand, large caliber self-propelled artillery carry only a limited supply of onboard rounds and rely on ammunition carried by tactical cargo vehicles for their main supply.

The method of ammunition stowage differs with the type of military vehicle. Combat vehicles that normally are not subjected to enemy direct fire, such as self-propelled artillery, stow their ammunition in any portion of the vehicle that is convenient to the weapon and to the loader. Vehicles that are expected to encounter direct fire, such as tanks, however, stow the main portion of their ammunition in low vulnerability areas, such as the lower portion of the hull, particularly toward the rear of the vehicle. Studies of ballistic hits on tanks indicate that the turret receives the largest portion of hits, based on the relative areas exposed. The hull area below the sponsons is generally a safer ammunition stowage location than one at a higher level. Tanks normally have some limited ammunition stowage facilities in the turret basket for ready rounds.

The design of ammunition stowage facilities should provide a secure rack or compartment in which to house the ammunition. Clearance should be provided between the rounds and the vehicle structure such that hull deformations caused by ballistic impacts will not contact the

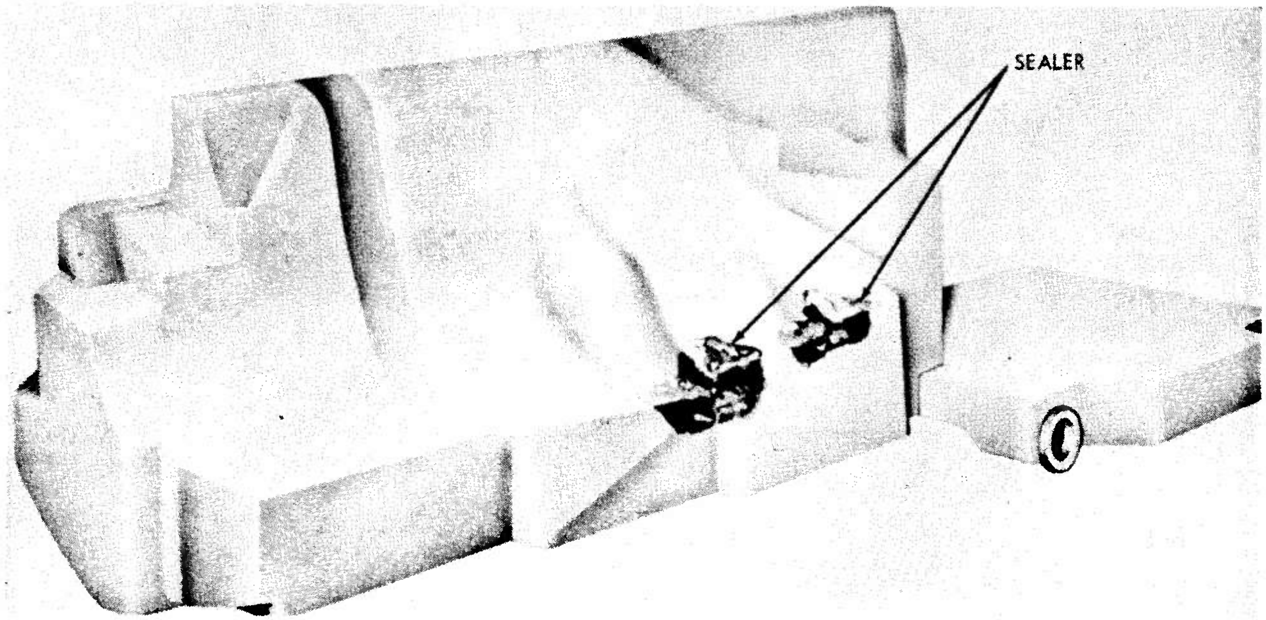


Figure 3-36. Left-hand Fuel Tank in M60 Combat Tank

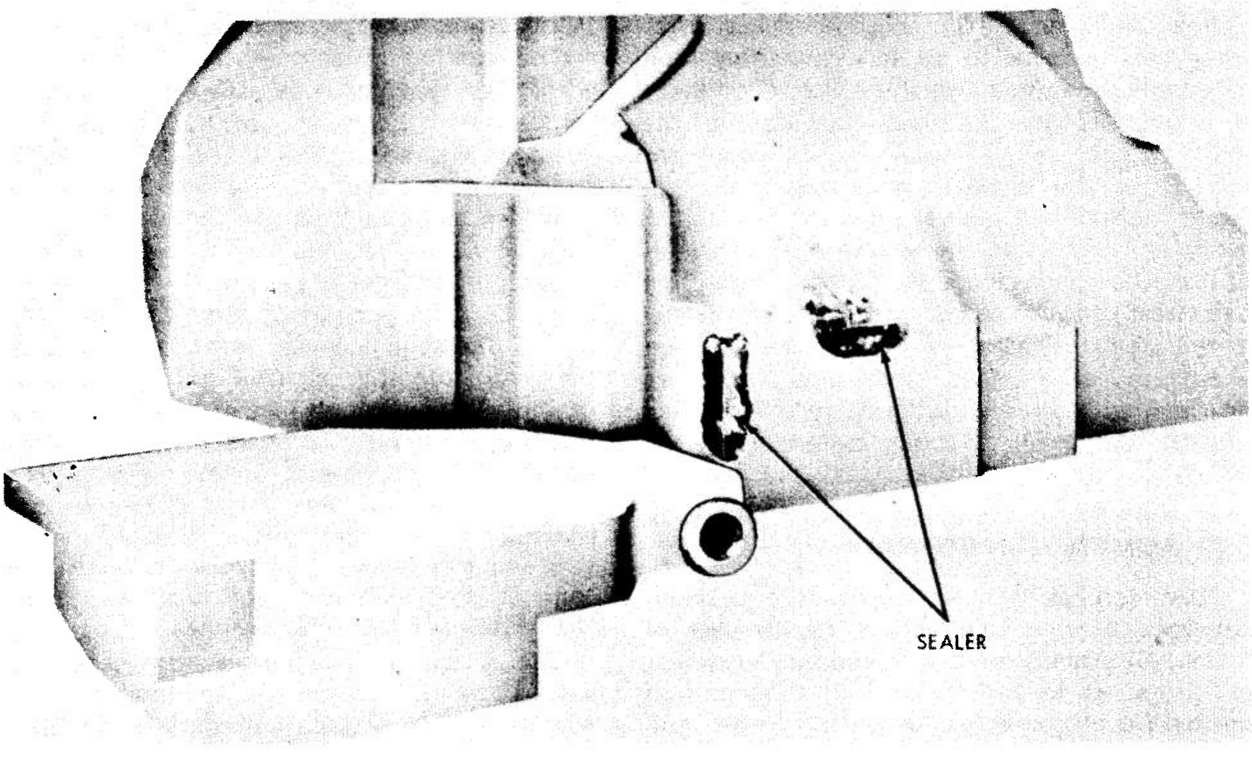


Figure 3-37. Right-hand Fuel Tank in M60 Combat Tank

ammunition. The ammunition should be so stowed that road shocks and ballistic shocks experienced by the vehicle will not result in concentrated forces against the primer end or fuze end of the ammunition.

Most ammunition, particularly the large caliber types, is stowed horizontally in various racks, trays, and boxes. Vertical racks are also used, however, in order to make full use of all available space. A fair number of ready rounds can be stowed vertically around the periphery of the turret basket of a combat tank without seriously crowding the fighting compartment. Furthermore, certain special types of ammunition, such as white phosphorus rounds, are required to be stowed vertically with their bases downward, particularly in hot climates. Thus, a number of vertical stowage racks are desirable for this additional reason.

Most horizontal stowage racks for large caliber ammunition (75 mm-105 mm) consist of a number of metal tubes, of a size to accept the specified caliber ammunition, welded to suitable supporting plates. The nose ends of the tubes are closed-off to prevent the rounds from sliding through. The base ends of the tubes are provided with spring-loaded hinged retainers across their openings to hold the rounds in place. The racks should be located and oriented inside the vehicle in the most advantageous position with respect to the loader and to the weapon which they serve; this is particularly significant with respect to the ready rounds. The rate of fire of a vehicle-mounted weapon depends upon the speed with which the loader can remove the rounds from their racks and load them, together with the ease with which he can identify the rounds as to type while they are in the racks. Rounds stowed outside of the turret basket and beneath deck plates are proportionately more difficult to handle. The functioning of catches and releasing devices, together with their accessibility, are also important considerations. Figs. 3-38 and 3-39 show various caliber ammunition stowage racks and boxes in the turret of the M60 Combat Tank.

3-40 PINTLES, TOW LUGS, LIFTING EYES, AND TRAVEL LOCKS

3-40.1 PINTLES AND LUNETTES

A pintle assembly (Fig. 3-40) is a towing hook equipped with a hinged latch across its opening to retain the eye of a tow bar, or similar item placed in the hook, from becoming inadvertently disengaged. A locking pin, thrust through the sides of the latch and hook, prevents the latch from opening accidentally. All pintle assemblies are of a rotatable type, i.e., they are made to swivel about their longitudinal axis, except those used on ¼-, ½-, and ¾-ton 4 X 4 trucks. These vehicles are generally equipped with nonrotatable pintle assemblies to meet a definite military requirement—keeping small, narrow-tread, towed loads from overturning in rough cross-country operations. Pintle assemblies are normally mounted at the center of the rear end of the vehicle in a manner that will transfer the towed load directly to the vehicle frame or equivalent structure. Generally, all military vehicles are equipped with a towing pintle at their rear; some special-purpose vehicles also have pintles at their fronts.

The heights above the ground at which pintle assemblies are mounted have not been standardized, as this dimension depends upon the size and general configuration of each vehicle. However, pintle heights should be compatible with the heights of the towing lugs or lunettes of the vehicles that are to be towed. Pintle assemblies, however, have been standardized. They are available in three capacities—18,000; 40,000; and 100,000 lb. Dimensions, physical characteristics, and other pertinent information relating to these assemblies are given in MS-51335, MS-51118, and MS-51117, respectively, of Ref. 69.

Pintles should be positioned to allow the most favorable draw line for towing trailers and other vehicles. The mounting must be sufficiently strong to withstand the shock loads associated with start up and with operations over rough terrain. The pintle location should be free of interfering brackets, braces, or body structure that can hamper coupling or uncoupling operations or limit relative motions between the towing and the towed vehicles. On some vehicles, particularly wheeled vehicles, a two-sectioned rear bumper is often provided, and the pintle assembly is located between the two sections. The bumper protects the vehicle frame from damage while the curved bumper surfaces guide the trailer lunette into the pintle assembly to facilitate coupling.

A lunette (also known as a drawbar

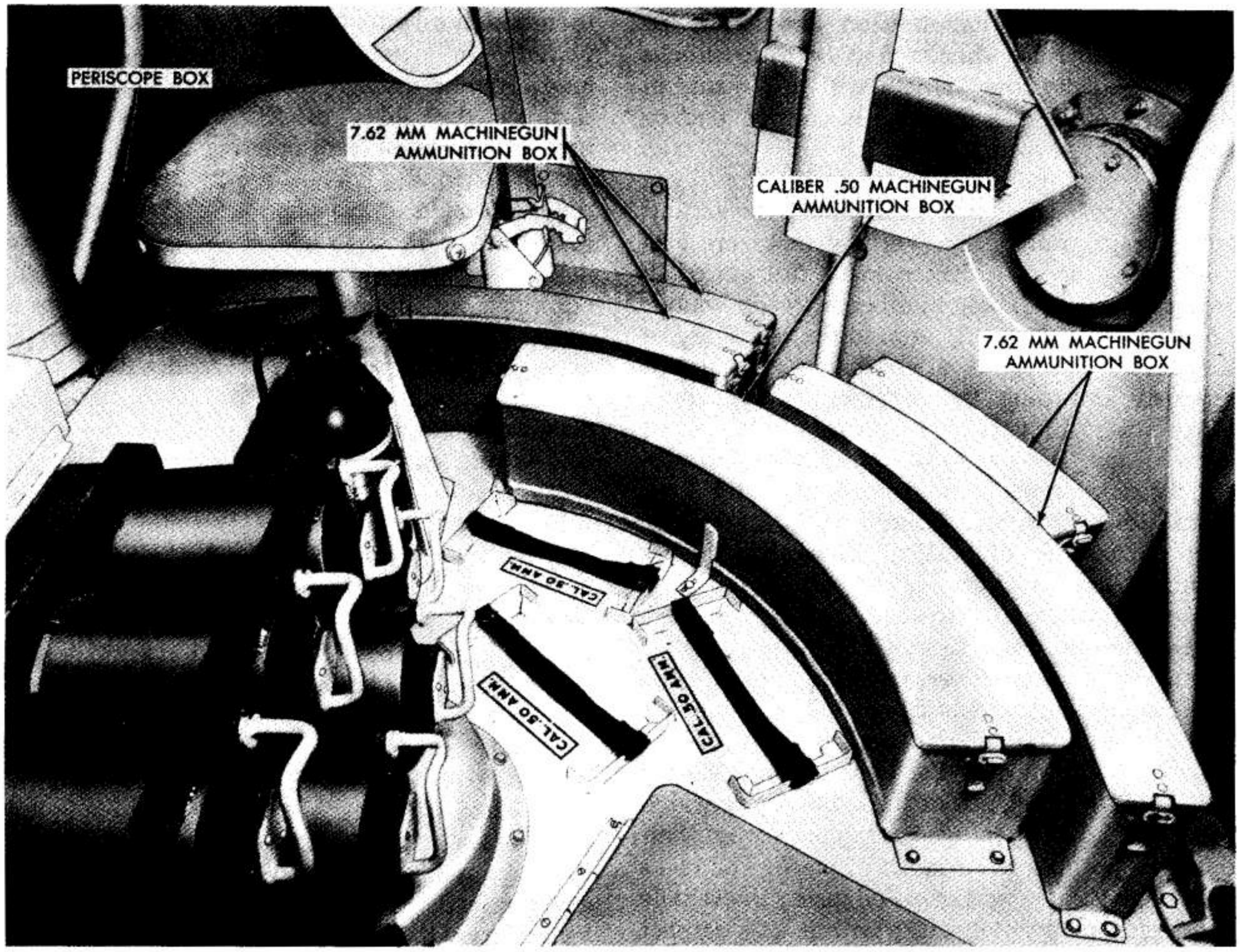


Figure 3-38. Ammunition Stowage Facilities in M60 Tank Turret

ring-coupler) is a towing ring or eye in the tongue or trail plate of a towed vehicle, such as a trailer or a gun carriage, that is used in attaching the towed vehicle to the towing vehicle (Fig. 3-40). The dimensions of the lunette eye have been standardized by MS-51336 of Ref. 69 to an eye with an inside diameter of 3 in. and a material cross-sectional diameter of 1-5/8 in. Generally, the lunette is allowed to swivel about its longitudinal axis, to permit the towing and towed vehicles to roll independently of each other; and it is generally spring-loaded axially to cushion shock loads somewhat. The dimensions and characteristics of standard heavy duty and light duty lunettes are given in MS-51337 and MS-51338 of Ref. 69, respectively.

For some applications, the capacity to be towed by a variety of vehicles is desirable. In order to accommodate the different pintle heights of the various towing vehicles, the towed vehicle is provided with an off-set lunette—the off-set being in the vertical plane. This design affords accommodation for two distinct pintle heights by rotating the lunette through 180°. By selecting the setting closest to the desired pintle height, a variety of towing vehicles can be used. The shank of the off-set lunette is kept from rotating inadvertently by a locking pin thrust through the side of the housing. Standard off-set lunettes are described in MS-51339 of Ref. 69.

3-40.2 TOWING HOOKS, SHACKLES, AND LUGS

In order to facilitate the towing of disabled self-propelled military vehicles, they are provided with towing hooks, shackles, or towing lugs at their front and rear. These are mounted to the frame side members, or to an equivalent load-bearing structure in hull-type vehicles. They can be seen on the various vehicle illustrations in Chapters 1, 2, and 3. In general, they are

Figure 3-39. Stowage Facilities in M60 Tank Turret

mounted at the same height above the ground as is the pintle assembly so that the same draw lines are achieved when towing by either the pintle or the towing hooks (or shackles or lugs). The pintle assembly is used only for towing unpowered trailer-type vehicles and should never be used for towing disabled self-propelled vehicles; the towing hooks or lugs are inteded for this latter purpose.

Combat tanks, self-proplled artillery, and similar heavy vehicles are generally provided with towing hooks (Fig. 3-41) rather than towing shackles or lugs. Towing hooks are used in conjunction with flexible steel cables that terminate in steel eyes at both ends (Fig. 1-27). Each vehicle equipped with towing hooks is provided with one such towing cable. Thus, the cables from both, the towing and the towed, vehicles are required. The towing cables must not be connected to the towing hooks by any means other than by the cable eyes, as doubling the cable results in short radius bends which cause the wire strands to break and leaves the cable weak and dangerous to handle. The towing cable should never be connected to the pintle.

Wheeled vehicles are generally provided with towing shackles (Figs. 1-10 and 1-11) rather than towing hooks. These are U-shaped metal forgings fastened to their supporting brackets by means of pins or bolts that pass through the legs of the U and about which the shackle can swivel in the vertical plane. Towing is accomplished by means of tow chains and hooks passed through the shackles.

Towing lugs are roughly triangular metal projections or brackets that are welded or bolted directly to a vehicle hull. They are usually mounted in pairs, one on each side of the

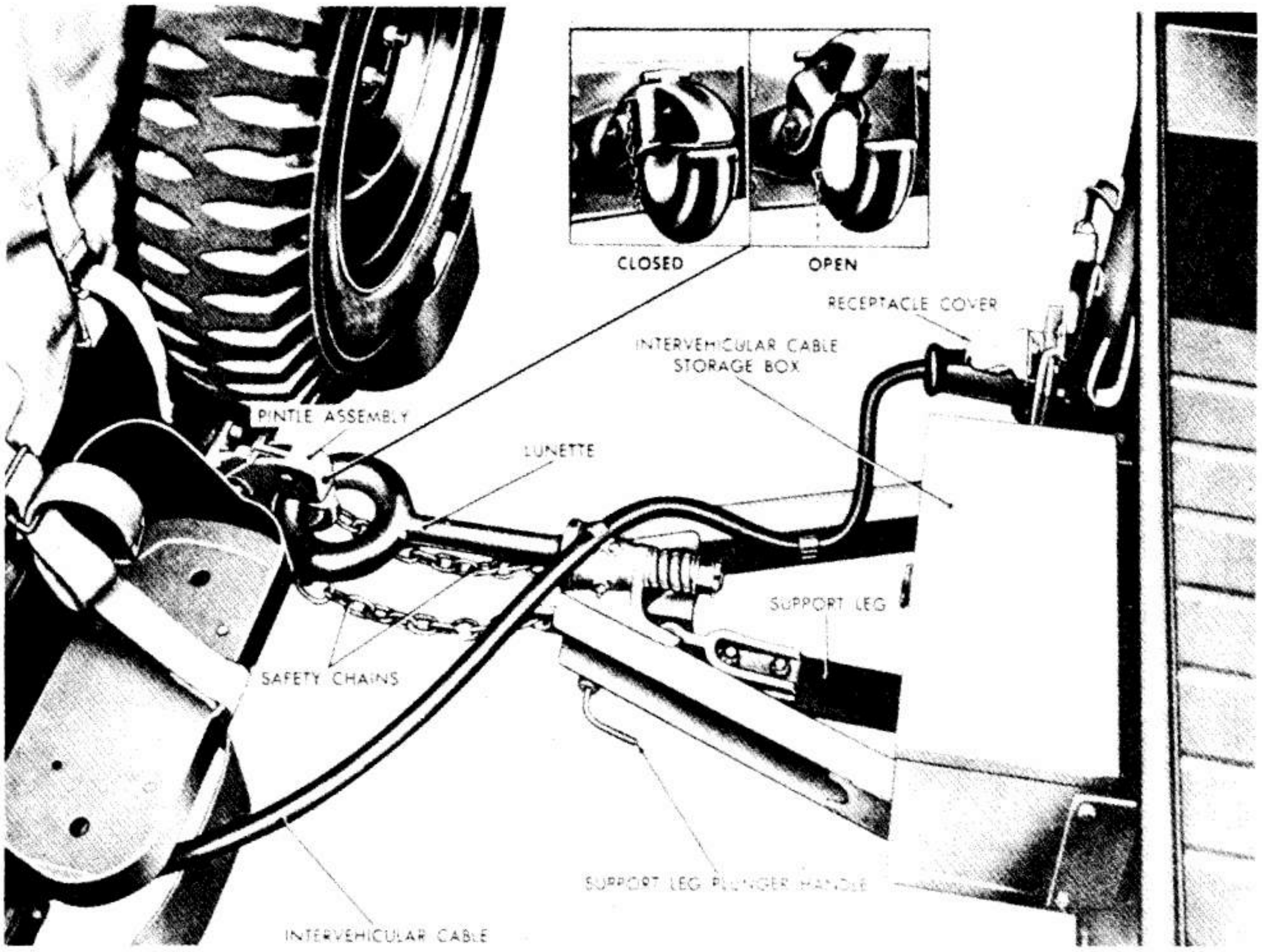


Figure 3-40. Pintle and Lunette Assemblies

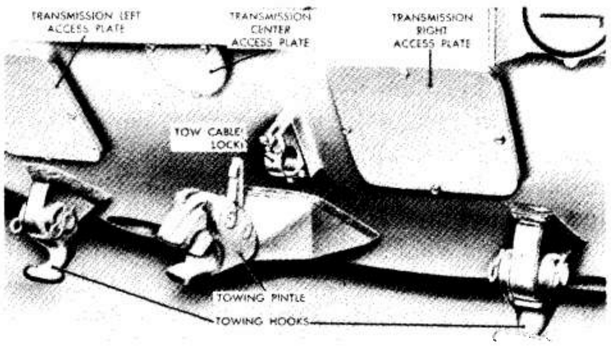


Figure 3-41. Towing Hooks and Pintle Assembly at Rear of Combat Tank

vehicle, at the vehicle front and rear in a manner similar to the locating of towing hooks. A round hole in each lug provides a means of attaching tow chains or hooks. Fig. 3-42 shows a vehicle equipped with towing lugs. Towing lugs of this type are customarily provided on medium and lightweight hull-type vehicles. A list of general requirements for the location of lugs and eyes on military vehicles is given in the paragraph which follows.

3-40.3 LIFTING EYES[70]

Military vehicles are generally transported from their place of manufacture to the place where they will be used, rather than being driven under their own power. They are also transported whenever the distances to be traveled are great. In the course of this transport, it is often necessary to lift or hoist the vehicles aboard various carriers by means of cranes or similar equipment. To facilitate these hoisting operations and to minimize the risk of damage to the vehicles that these types of operations often entail, suitable lifting eyes (slinging eyes) must be provided as integral parts of each vehicle. These may be in the form of

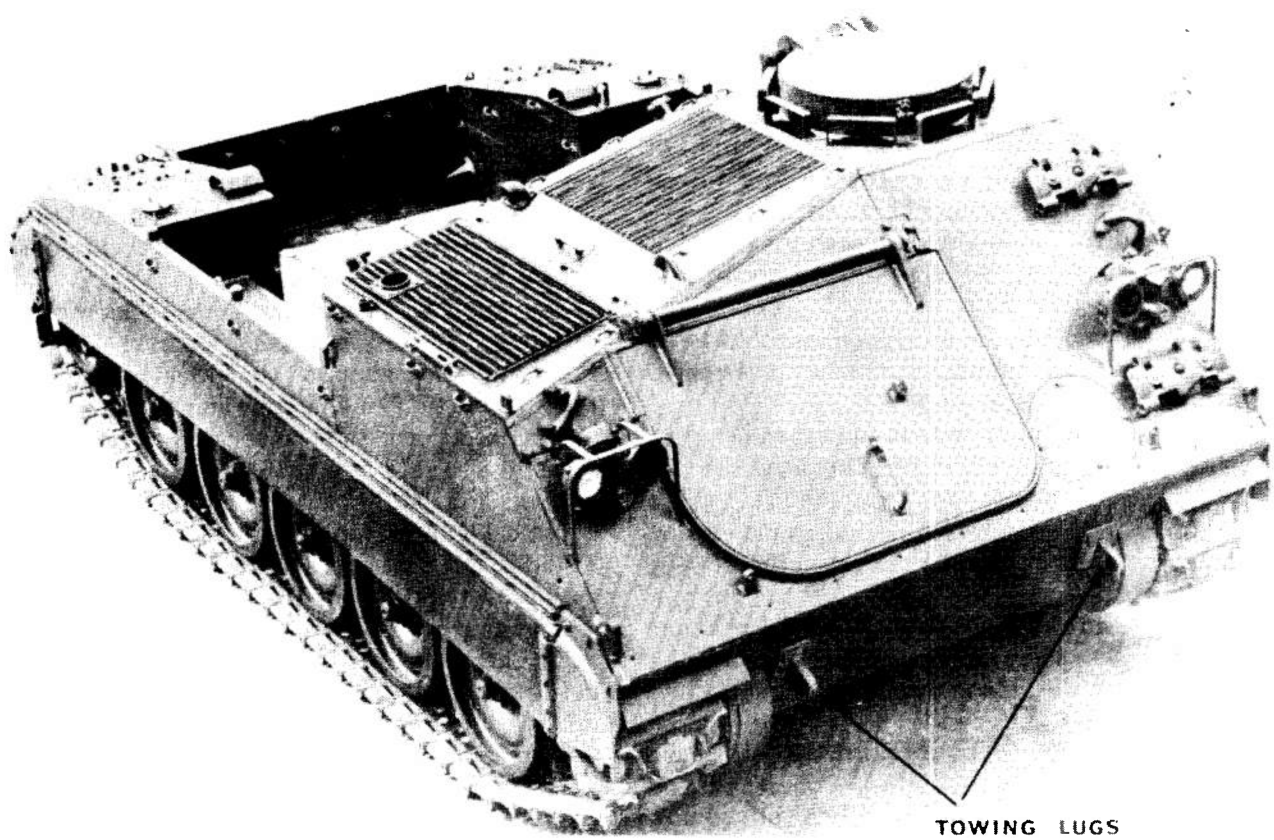


Figure 3-42. Towing Lugs on High-speed Tractor

lugs or pad eyes, and may include a shackle or a ring as an integral part. Their function is to provide a means of attaching a hook, sling eye, or shackle to the vehicle to permit safe lifting.

Good design practice is to provide a minimum of four points for lifting on all vehicles. In order to minimize the total number of attachment eyes on a vehicle, the locations and designs of lifting eyes, eyes for tying down the vehicle, and towing lugs should be coordinated for multipurpose use wherever this is practicable. The following list of requirements is applicable to the designs and location of all types of attachment eyes (lifting, tiedown, and towing):

a. Adequate clearance must be provided between attached lugs and other exterior parts of the vehicle (see Table 3-14).

b. Attachment lugs, eyes, and related devices must not interfere with the functioning of the vehicle.

c. Attachment lugs, eyes, etc., must be located so as not to increase the rectangular prism required by the vehicle in the hold of a cargo carrier.

d. Lugs and eyes must be located for maximum accessibility to them.

e. The material used for all attachment lugs and eyes should have an ultimate tensile strength of not less than 60,000 psi and a minimum yield strength of 30,000 psi.

f. The working load upon which the designs of all lugs and eyes should be based is the maximum resultant static load imposed on the eye under the service conditions anticipated and considered with the leg of the attached lifting or tiedown device acting at an angle of 45° from the vertical.

g. The working stress upon which the designs of all lugs and eyes should be based is the maximum combined stress that is developed in the lug or eye by the application of the working load just defined.

h. The ratio of the ultimate stress of the material used for the lugs or eyes to the working stress just defined (factor of safety) should be a minimum of 5.

The location of lifting, or slinging, eyes should meet the following requirements:

a. Sling legs attached to lifting eyes should be able to converge over the center of gravity of the fully loaded vehicle as it is prepared for shipment.

b. The axes of the attached sling should not exceed a height of 19 ft above the lowest extremity of the vehicle.

c. The lifting eyes should be so located that the apex angles of the sling legs do not exceed 45° from the vertical.

d. The plane of the lifting eyes should be in approximate alignment with the anticipated angle of the sling legs.

e. Lifting eye locations should be selected so as to minimize, or eliminate, the necessity for using spreader bars when making the lift.

f. The lifting points selected must assure stability of the vehicle during the lifting operation.

g. When hub attachments are used for lifting wheeled vehicles, they should permit the alignment of the lifting eyes with the sling legs to prevent torsional stresses.

h. The working load apportioned to each lifting eye as it is attached to the vehicle, based upon the 45° angularity specified previously, should be 35 percent of the total weight of the equipment when in its maximum loaded condition as prepared for shipment[70].

Recommended dimensions of lifting and tiedown eyes for wheeled and tracked vehicles are given in Table 3-14.

3-40.4 GUN TRAVELING LOCKS

Gun traveling locks are provided on all tanks and self-propelled guns. Their function is to support the gun near its muzzle end, when the vehicle is traveling in a noncombat situation, to prevent unnecessary barrel whipping due to road shocks, and to limit the resultant shock loading that would otherwise be transmitted to the gun elevating mechanism. Traveling locks are mounted on either the front or rear top deck of the vehicle hull. The choice depends upon the general vehicle configuration and the length of the gun barrel.

Tracked combat vehicles in which the power plants are located in the forward portions of the hulls generally have their turret centers mounted somewhat aft of the hull center. This configuration results in less weapon overhang (projection of the weapon beyond the hull) in the front of the vehicle. Traveling locks placed at the fronts of these vehicles will be closer to the gun muzzle and will, therefore, be more effective than if placed at the vehicle rear. On the other hand, the reverse is true in vehicle configurations in which the power plant is in the rear of the vehicle. In these cases the traveling locks are mounted on the rear deck, and the gun is traversed to the rear when it is to be locked for traveling.

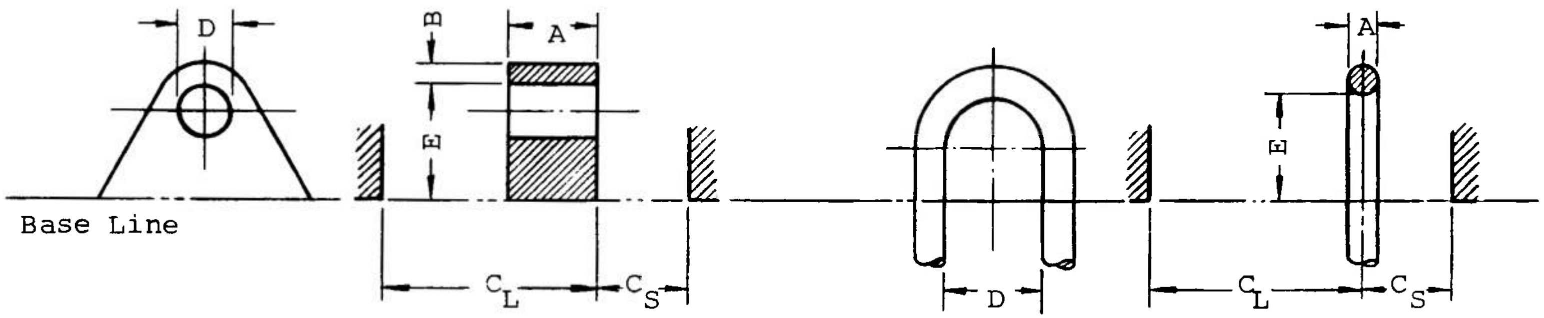
The designs of gun traveling locks vary on different vehicles, but they all have the same function. They support, and lock in place, the forward portion of the gun, and they must fold down flat against the top deck when not needed so as not to interfere with the traversing of the weapon. Fig. 3-43 shows the component parts of a typical gun traveling lock. The gun barrel is rested in the top saddle of the bracket, the cap is swung over the top of the barrel, and the barrel is clamped in place by screwing down the lever bolt. The lever serves as a handle in turning the bolt and folds down over the cap when not needed. Other illustrations of gun traveling locks can be seen, both erected and collapsed, in Figs. 1-29, 2-10, 3-25, and 3-26.

3-41 TIEDOWNS[70-72,74]

Tiedowns, as the term is used here, are hardware items mounted on all military vehicles for the purpose of providing a means of attaching lashings, chains, cables, or other restraining devices to the vehicle. They are required for securing the vehicle on board carriers during vehicle shipment by road, rail, water, or air; they are required for securing cargo to the vehicle; and they are required for securing canvas protective covers over the cargo holds of open vehicles. Tiedown hardware is generally in the form of lugs, rings, eyes, shackles, and cleats that are fastened to the vehicle at key points by bolts, screws, or welding. Refs. 70 to 72 give detailed specifications on the design and positioning of tiedown devices. The general requirements for lifting eyes given in par 3-40.3 are also applicable here.

A minimum of four points should be provided for securing a vehicle on board a carrier. These must be positioned to keep the vehicle stable against vertical loads as well as loads in the longitudinal and lateral directions. Dimensions of tiedown eyes should conform to those given

TABLE 3-14 RECOMMENDED DIMENSIONS OF LIFTING AND TIEDOWN EYES[70]



Class	Weight Range, Long Tons (2,240 lb)	A*		B*		C_L *‡	C_S *‡	D*		E*	
		max	*min*	*max*	*min*	*min*	*min*	*max*	*min*	*max*	*min*
		For Tanks and Track-laying Vehicles									
1	1 to 10	1-1/2	7/8	1-1/8	7/8	9	4	3-1/2	3	5	3
2	10 to 22	2	1	1-1/2	1	12	5	3-1/2	3	5	3-1/2
3	22 to 45	2	1	1-1/2	1	16	7	4	3-1/2	6	5
4	45 to 65	2-1/2	1-1/4	2	1-1/4	20	8	5	4	6	5-1/2
		For Wheeled Vehicles									
1	1 to 5	1	3/4	1	3/4	7	3	3-1/2	3	5	3
2	5 to 10	1-1/2	7/8	1-1/8	7/8	9	4	3-1/2	3	5	3
3	10 to 22	2	1	1-1/2	1	12	5	3-1/2	3	5	3-1/2
4	22 to 45	2	1	1-1/2	1	16	7	4	3-1/2	6	5
5	45 to 65	2-1/2	1-1/4	2	1-1/4	20	8	5	4	6	5-1/2

*All dimensions are in inches.

‡ There shall be no interference within C_L and C_S that could interfere with the engagement of a shackle or pin with the eye.

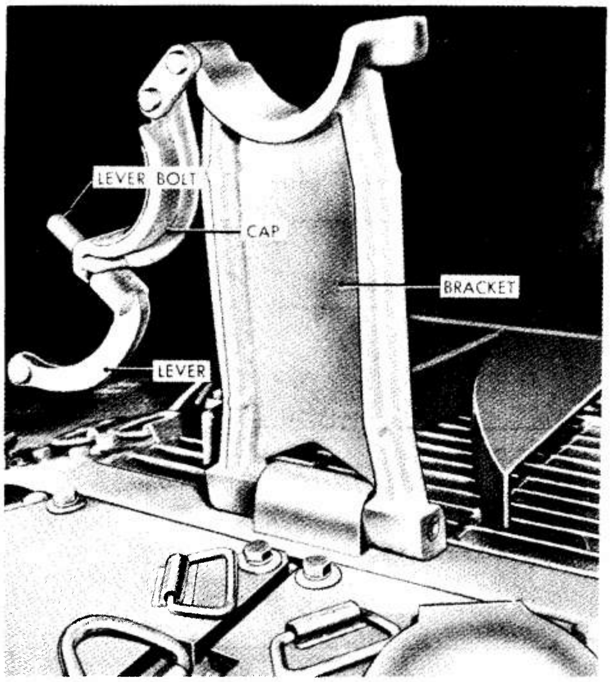


Figure 3-43. Typical Gun Traveling Lock

in Table 3-14. Where practicable, eyes should be designed for multipurpose use, such as for lifting and for tying down.

The working load (see par. 3-40.3 for definition) apportioned to each tiedown or multipurpose eye as attached to the vehicle should be 70 percent of the maximum total restraint load anticipated during shipment, or 70 percent of the actual weight of the loaded vehicle as prepared for shipment, whichever is the greater.

Lifting eyes are often attached to the hubs of wheeled vehicles. These may be used to supplement the vehicle tiedowns, but they should not be used as principal tiedown points.

Provisions should also be made for restraining cargo in and on vehicles—particularly heavy cargo such as tractors, tanks, heavy weapons, and other vehicles when loaded on board trailers, and partial loads of crated material or heavy equipment in cargo carriers. The cargo in partially loaded vehicles has a tendency to shift during vehicle accelerations. This shifting can take place in any direction—longitudinal, lateral, and vertical—and if not prevented by proper blocking and tiedown restraint, can result in disaster to the vehicle. Cargo tiedowns in the form of rings and cleats should, therefore, be provided on the vehicle floor, walls, or fastened to frame members at suitable locations. In the cargo compartment, these tiedown fittings should be recessed to minimize interference with cargo loading and unloading operations.

Vehicles with open cargo compartments are generally supplied with canvas covers to protect the cargo from inclement weather. Provisions should be made on these vehicles for securing this canvas. Rings and cleats, evenly spaced on the exterior side panels of the cargo body, serve this purpose effectively (Figs. 1-9 and 1-31). When bows are used to support the canvas cover, one tiedown is generally provided on the exterior side of each bow leg and additional ones are placed across the front and rear. Cover tiedowns should normally not be placed on doors or gates as the tiedown lashings will interfere with the functioning of these closures. Refs. 71 and 72 are an excellent source for detailed information on tiedown devices and tiedown procedures.

3-42 HINGES, LATCHES, AND HANDLES[73]

Ref. 73 describes vehicular door hardware for use on van-type tactical vehicles. The items described are commercially available and have been found acceptable for military applications. No comparable document is available for door hardware for other types of tactical or combat vehicles, therefore, the design of hinges, latches, and handles for other vehicles is left to the particular application and to the ingenuity of the designer.

Hinges vary in size and complexity from simple piano-type hinges used on access doors to armored hinges used on hatches of armored combat vehicles. Continuous-type hinges have been standardized and are described in MS-20001 and MS-35000 of Ref. 69. These hinges are available in various thicknesses, lengths, and materials and are useful for access door or panel applications. Hinges used for crew or cargo doors, for hatches, and for gates are generally of the automotive type, which are capable of carrying greater loads than are piano-type hinges. Automotive-type hinges are cold-formed, cast, or forged depending upon their shape and strength requirements.

Latches are required for all doors and hatch closures. Access doors that are lightly loaded require only simple latches which may be of the self-closing and quick-opening types. Where quick-opening latches are used, they must be

capable of being opened readily by a person wearing heavy gloves. When arctic service is specified or anticipated for a particular vehicle, latches must be operable by personnel wearing heavy arctic mittens. Automotive type latches are useful for many tactical vehicle applications, particularly to secure personnel doors. Latches on doors and hatches of armored combat vehicles should be of the positive locking type, similar to that shown in Fig. 3-3, and should be provided with means of locking from the inside. In addition, the outside of one hatch should be provided with a means for attaching a padlock. In designing latch or hinge installations for armored combat vehicles, it is important to design the door mounting in such a manner that neither the latch nor the hinge will be subjected to ballistic impact loads. These loads should be transmitted from the door or hatch closure directly to the door support structure, which is usually the hull.

Handles are required on doors and hatches (usually in connection with latches) and on various parts of the vehicle to facilitate personnel access and to provide hand holds to the passengers and crew when the vehicle is crossing rough terrain. Access handles are also placed on the exterior sides of vehicle cabs, to serve as grab handles for the crew when boarding (Fig. 1-10) and on, or near, tailgates to assist personnel in climbing into the cargo holds. These handles are often installed in conjunction with boarding steps.

3-43 EXTERIOR STOWAGE RACKS

Exterior stowage racks are used to provide stowage facilities for equipment and material that are not used inside the vehicle or that need not be immediately accessible for the performance of the vehicle's mission. Typical items commonly stowed in exterior racks are towing cables, tools, spare parts, maintenance equipment and supplies, pioneer tools, spare tires or track shoes, fuel cans, water cans, ladders (on some van-type vehicles), cleaning rods for tanks and self-propelled guns, spare bows and tarpaulins, camouflage nets, rations, and personal equipment such as duffle bags. Fig. 2-24 shows some of the equipment stowed on the outside of a large self-propelled gun, and Figs. 3-4 and 3-26 show some of the racks and stowage boxes on the exterior of combat tanks.

Three general rules should be observed concerning exterior stowage—(1) the amount of exterior stowage should be kept to a minimum, (2) the stowage racks and stowed items should not interfere with the functioning of the vehicle, the weapon system, or with any other functional equipment mounted on the vehicle, and (3) the exterior stowage should be so located on armored vehicles that the effectiveness of the armor protection will be compromised as little as necessary. The last consideration is particularly important in the design of direct fire-type armored vehicles since every bracket and rack attached to the armor decreases the level of protection at the point of attachment. For this reason, exterior equipment stowage on tanks should, where possible, be confined to positions on the fenders, at the rear of the turret (bustle), and at the rear of the vehicle.

Standard brackets are available for fuel and water cans, and for the mounting of pioneer tools. These are described in Ref. 69. Racks and brackets for other exterior stowage require special considerations peculiar to the type of equipment to be stowed and to the vehicle on which it is to be mounted. Straps or latches are required to secure the stowed equipment within the racks. All outside stowage brackets, racks, and boxes must be designed to withstand the abuse of brush, tree limbs, thrown rocks, sand, mud, and blast. In addition, under adverse weather conditions—rain, snow, sleet, freezing conditions, or sandstorms—none of the externally stowed items should become frozen in or otherwise made inaccessible. Furthermore, under stress of combat or an accident, perhaps with the vehicle overturned, or on its side, emergency items should still be accessible.

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CHAPTER 4
DESIGN PROCEDURES AND DESIGN CRITERIA*

SECTION I—STRUCTURAL ANALYSIS

4-1 FUNDAMENTALS OF DESIGN

4-1.1 GENERAL

In the preparation of this chapter, it is assumed that engineers using this handbook are throughly familiar with the basic principles of strength of materials such as can be found in many standard textbooks on the subject. For those who wish to review these principles, any of Refs. 1 through 5, given at the end of this chapter, are recommended although numerous others exist that are equally good. Instead of repeating this material, it was decided to show how these principles can be applied to specific areas encountered in the design of military vehicles and to emphasize principles that have particular importance in this field. In addition, as an accurate assessment of the magnitudes of the loads or forces acting on a system must be the prelude to a successful system design, it was decided to give some guidance in making these load assessments.

The designer of a structure or of a mechanical system is concerned with the integration of three principal objectives; namely, (1) a logical and efficient organization of space and allowable weight limitations for their maximum utilization in achieving specified functional goals; (2) an arrangement of masses, motions, and inter-related forces to produce desired kinematic and dynamic results; and (3) the detailed specification of all structural and functional components as developed from an evaluation of their strengths and elastic characteristics when subjected to the loads and forces anticipated for the system. The attainment of each of these goals requires considerable knowledge, experience, and ingenuity; but the attainment of objective (3) is often the most difficult to achieve accurately because of the difficulty in determining the true magnitudes and characteristics of the forces to which the various elements will be subjected. These are the so-called applied loads, and upon their accurate evaluation rests the efficiency of the design. If they are underestimated, the system will fail; if they are overestimated—or if they are increased unrealistically through the application of ignorance factors—the system will suffer due to excessive size and weight, a poor utilization of space, functional impairment, increased cost, and the general design inefficiencies normally associated with overdesign. Later in this section, the sources and magnitudes of some of the loads to which military vehicles, particularly their bodies and hulls, are subjected will be discussed. Before proceeding, however, the principles given in the following paragraphs should be reviewed, as they are fundamental to the satisfactory design of mechanical and structural components that are subject to static and dynamic forces.

The design of machine parts may be developed from either a strength (stress) basis or from a stiffness (strain) basis, but the choice is not an arbitrary one. In some cases, the strain produced is the more important factor.

When strength alone is the basis of design, the parts are proportioned to be sufficiently strong to withstand the stresses developed in them by the applied loads; and as long as no rupture occurs, the parts designed in this manner fulfill their intended purpose. As an illustration, consider a tow chain used for towing heavy vehicles. This tow chain will fulfill its function of towing vehicles regardless of the amount of stretch that occurs in the chain, provided the load does not produce a stress sufficiently large to rupture the chain. That is, the stress developed in the chain by the maximum applied load is the governing factor in the design of a tow chain. Any measures taken to minimize the stretch of the chain are unnecessary in this case, and will only lead to an overdesign.

When strain or stiffness is the basis for design, the elastic behavior of the components under the maximum applied load must be investigated.

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The parts must then be proportioned to maintain their distortions within acceptable limits—limits that will permit the parts to perform their prescribed functions. Stiffness, or rigidity, is often more important than strength alone. Excessive distortions of a vehicle frame or body may cause doors to jam, impair steering control, cause malfunctions of operating equipment that may be mounted on the vehicle, and generate unnecessary vibrations and noise. Similarly, rigidity of mounting with respect to a fixed reference is essential to the accurate performance of sighting and aiming and fire control devices on board combat vehicles. In these cases, merely designing the parts to be strong enough not to rupture under the applied loads is not enough. The parts must be sufficiently rigid to hold their deflections within acceptable limits for adequate functional performance.

Furthermore, when strain is the basis for design, two types of strain must be considered—elastic strain and inelastic (or plastic) strain. Loadings below the yield strength or proportional limit of the material produce elastic strain; i.e., the material so deformed will return to its original size, shape, or position when the load is removed. Loadings above the yield strength or proportional limit, however, produce a certain amount of inelastic, or permanent, deformation. The amount of permanent deformation depends upon the degree to which the yield strength was exceeded and upon the physical characteristics of the material.

The mere fact that the loaded member experiences permanent deformation, however, does not always signify the failure of a part; functional impairment should be the criterion of failure. Many components—such as lifting eyes, towing lugs, bump stops, noncritical brackets, air extraction and airdrop components, spades, etc.—experience no significant functional impairment when partially deformed. The additional material (and attendent increased weight, space, and cost) involved in eliminating all possibility of any permanent deformation in such components is not justified. This is especially true when the maximum applied load upon which the design is based is in the realm of the possible but highly improbable. For example: a designer may conceive an extreme situation in which three legs of a four-legged hoisting sling become separated from the load during a hoisting operation; and, in order to be safe even in this situation, he may wish to make each lifting eye capable of supporting the entire weight of the load. Since this situation is so highly improbable, the risk of incurring permanent deformation of the lifting eye, in the event this situation did occur, is entirely acceptable, as long as the eye fulfilled its function of restraining the load. Undue avoidance of the possibility of permanent deformation leads to overdesign, excess weight, the need for more power in a vehicle, and increased costs. Therefore, the conditions of functional failure of every part should be realistically determined before establishing its strength limits. 4-1.2

4-1.2 TERMINOLOGY

4-1.2.1 Stress and Strain

The terms *stress* or *unit stress* as used in this chapter signify a force or load per unit area of the loaded element and is a measure of the intensity of the force acting on a definite plane passing through a given point. The distribution of the stress may or may not be uniform, depending upon the nature of the loading conditions. Tensile stresses as found from the typical equation $S_T = P/A$ are considered to be uniformly distributed; whereas tensile stresses due to bending as determined by means of the equation $S_{TB} = My/I$, refer to the stress at a point located at a distance y from the neutral axis of the loaded member. Therefore, the stress over the cross section of a member subjected to bending is not uniform. Similar situations exist for compressive and shear stresses.

Stresses may act perpendicularly or parallel to the plane under consideration. Those which act perpendicularly to the plane are the *normal* stresses and those which act in a parallel direction are the *shear* stresses. Normal stresses may be either tensile or compressive stresses according to whether they tend to pull the adjacent particles of the material apart *(tensile stresses)* or press them together *(compressive stresses)*.

If three mutually perpendicular planes are passed through a stressed point in a material and oriented in such a manner that only normal stresses exist on the planes, all other stresses being zero, these normal stresses are known as the

principal stresses. The numerically largest of these is the *maximum principal stress*.

The term *strain* refers to the change in length of a member caused by a load to which it is subjected, and unless specified otherwise, implies a change in length per unit of original length of the loaded portion of the member. Strain distribution may or may not be uniform in a complex structure, as is the case with stresses, and depends upon the nature of the loading.

Strains also occur in directions other than those of the applied forces. A *normal strain* is that strain that is associated with a normal stress, and it occurs in the direction in which its associated stress acts. Normal strains can be either increases or decreases of original length according to the type of stress that produced them. *Principal strains*, like principal stresses, are normal strains that act on three mutually perpendicular planes which are so oriented that only normal strains exist, all shearing strains being zero.

When the loading condition on a member is uniaxial, i.e., in one direction only, the strain in the direction of the applied stress varies with the stress. The ratio of stress to strain has a constant value within the elastic range of the material but decreases when the elastic range is exceeded and plastic strain occurs. This constant ratio of stress to strain is known as the modulus of elasticity and is generally designated by the symbol E. Its numerical value is different for different materials and is generally different for any one material when subjected to different types of stress. Table 4-1 lists the moduli of elasticity of serveral common materials.

Axial strain is always accompanied by lateral strains of opposite sign in the two directions mutually perpendicular to the axial strain. Under uniaxial loading conditions, the ratio of either of the lateral strains to the axial strain is known as Poisson's ratio. For stresses within the elastic limit, this ratio is approximately constant. Values of Poisson's ratio for a number of common materials are shown in Table 4-1.

In multiaxial loading, the strains that result from each of the applied stresses are additive. Therefore, the strains in each of the principal directions must be evaluated, taking into account all of the principal stresses and Poisson's ratio. The resultant strain can then be determined by adding the principal strains vectorially.

4-1.2.2 Standard Load Nomenclature

4-1.2.2.1 Applied Loads

The term *applied load*, or simply *the load* is the actual external force that acts upon the component under consideration. However, when a study is made of the loads that may be experienced by a military vehicle being designed, it is usually found that it will experience loads of various magnitudes under different situations. The natural procedure under such circumstances is to determine the maximum load values that are ever apt to occur (and perhaps add a little for safety's sake) and develop a structure that will satisfactorily resist these loads. Unfortunately, this procedure will inevitably lead to overdesign. See par. 4-1.2.3.1 for a more complete discussion of this subject.

A design doctrine that has as its goal the capability of successfully meeting every eventuality is unnecessarily severe. Situations can be postulated that are within the realm of possibility but are known to have a very low incidence, or even to be highly improbable. In such cases it is preferable to take the risk that the improbable will not occur (or at worst will occur infrequently) and design for the less severe situation. The alternative will penalize the design unnecessarily by providing all units with the capability of surviving a situation that most of them will never encounter. According to this doctrine, some failures in service are expected and are acceptable, provided their consequences are not intolerable.

As an illustration: it is entirely conceivable that a dump truck could accidentally skid from a haul road and plunge to the bottom of a gravel pit, or that a malfunctioning parachute during an airdrop could result in the truck striking the ground at near terminal velocity. It would be folly to strive for a design that would resist these situations, because they do not represent the normal operating mode for the vehicle and the resulting design would be unacceptable for normal operations. These are extreme situations, but less extreme situations occur that require a more careful evaluation. The probability of a situation occurring should be determined

TABLE 4-1 AVERAGE VALUES OF THE MODULI OF ELASTICITY AND POISSON'S RATIO FOR COMMON MATERIALS

Material	*Moduli of Elasticity E*, lb/in.^{2}		
	Tensile and Compressive	*Shearing*	*Poisson's Ratio* μ
Aluminum Alloy, Rolled	10,000,000	3,800,000	0.330
Iron, Cast, Gray	15,000,000	6,000,000	0.255
Iron, Cast, Malleable	23,000,000	10,500,000	0.271
Iron, Wrought	27,000,000	10,000,000	0.278
Steel, Cast	28,500,000	11,300,000	0.265
Steel, Mild	30,000,000	12,000,000	0.287
Steel, Hard	30,000,000	12,000,000	0.295
Brass, Rolled	12,000,000	5,000,000	0.331
Bronze, Rolled	14,000,000	6,000,000	0.33
Concrete (1:2:4 Mix)	2,500,000	—	0.10
Timber, Yellow Pine	1,500,000	—	—
Timber, White Oak	2,000,000	—	—

statistically, and the total impact of meeting this situation should be evaluated in a quantitative way if possible. This will permit a logical assessment of the desirability and worth of designing to meet this particular condition.

The applied loads that should be considered first are the static loads experienced by the various structural elements of the vehicle. These loads are due to the dead weights of the components that make up the vehicle, to the weights of the cargo and crew, and to the weights of the structural elements themselves. Under operating conditions involving rapid movement of the vehicle over rough terrain, firing of heavy caliber weapons mounted on the vehicle, amphibious and airdrop operations, transportation of the loaded vehicle by rail, ship, or aircraft, and the operation of certain vehicle-mounted heavy equipment (piledrivers, shovels, draglines, cranes, lifting platforms, etc.), the once static masses of the vehicle acquire kinetic energy. Velocities and accelerations enter the picture increasing the loading conditions imposed upon the supporting structures by introducing dynamic conditions—i.e., impact loads, shocks, and vibrations. These dynamic loads are generally much more severe than the static loads and, therefore, are the limiting or maximum applied loads upon which the design should be based.

Realistic loading conditions, or situations that the vehicle is *expected* to sustain and continue operations with no significant loss of characteristics, should be the ones considered. For the sake of a reasonable design, it should be remembered that some nondetrimental permanent deformation of components is tolerable, particularly when resulting from an

uncommon loading condition. A bent headlight guard on a combat tank, caused by a direct impact with a large tree, does not prevent the tank from continuing its operation. It is not a common occurrence, and the guard can be easily repaired later.

Generally, items that are associated with the armament and suspension system are more sensitive to deformations than is the hull or body structure. Items that are usually sensitive to slight deformations should be considered for a redesign to reduce this sensitivity. Obtaining realistic values for the applied loads is the most difficult part of the designing process.

4-1.2.2.2 Design Load

The term *design load* is often misinterpreted by designers and the result is overdesign. Design load represents the theoretical value at which structural failure is expected; and structural failure may take any one of three forms—(1) an actual rupture of the material of the loaded part, (2) an inelastic failure and collapse such as experienced in the buckling of a column or an edge loaded plate, or (3) a permanent deformation of a critical component which prevents it from performing its intended function. Design loads are approached, or exceeded, as the result of accidents and unexpectedly severe loadings. They are risk loadings such as might result from striking large boulders, concrete walls, bad airdrops, or direct hits by high explosive projectiles. Design loads are also known as the ultimate loads or ultimate design loads. They are determined by multiplying the applied load by an appropriate safety factor; or to state this in the form of an equation

Design Load=Applied Load × Safety Factor.
The safety factor is discussed in par. 4-1.2.3.

4-1.2.2.3 Allowable Loads

The allowable load, or working load, for a part is the maximum load that is considered safe for that part to resist in service. It is determined by the shape and size of the loaded member, the loading conditions, and the mechanical properties of the material used. The principal mechanical properites of concern here are the allowable stresses of the materials used. Allowable stresses, also referred to as working stresses, are the maximum unit stresses that are considered safe for materials when the material is subjected to specified loads (tension, compression, shear) in service. Where simple loading conditions exist, i.e., where failure will be due to a failure of the material rather than to an inelastic collapse of the member as is the case when a column buckles, the allowable load is that load which produces a unit stress within the material equal to the allowable stress for that material. When the loading condition is such that failure will be due to an inelastic collapse (buckling, etc.), the allowable load is the maximum load that the member can resist.

Most designers prefer to consider the allowable or working stress rather than the allowable load. Allowable stresses are considerably lower than the ultimate strength of a material. For brittle materials (materials that do not have yield points), such as cast iron and high carbon steels, the allowable stress is generally some proportion of the ultimate strength of the material. For ductile materials, such as structural steel and wrought iron, the allowable stress is taken as either a certain proportion of the ultimate strength or a certain (but different) proportion of the yield-point strength. In general, the allowable stresses of ductile materials are less than the yield-point stresses, because most structures will not fulfill their functions satisfactorily if the structural members are stressed beyond their yield points and thus become permanently deformed—although they might be safe against rupture or collapse.

Recommended values of allowable stresses for various materials have been determined through extensive tests and accumulated experience obtained under service conditions. They take into account variations in mechanical characteristics of different samples of the same type of material, uniformity of internal structure of the material (blow holes, laminar fissures, shrinkage stresses, etc.) whether the material is ductile or brittle, and the type of loading involved (static or dynamic). Allowable stresses for duticle materials are relatively larger in relation to their ultimate strengths than for brittle materials. This is because ductile materials can yield if subjected to unexpected overloads of if high localized stresses occur at some portion of a loaded member—and such unexpected stresses do occur. The yielding of

the material distributes some of the excess stress to surrounding materials and, hence, tends to prevent the member from breaking. Furthermore, the allowable stress for any one material is generally smaller when the material is to resist impact or repeated loads than when the load condition is static. This is because the true magnitude of impact loads is generally more difficult to evaluate; they produce larger stresses and strains than do loads of the same magnitude applied gradually; and localized stresses and nonhomogeneity of the material have a much greater influence on the strength of a member subjected to repeated loads than when the same loads are applied statically.

Recommended values for allowable stresses have been tabulated and are published in materials handbooks, engineering handbooks, and other references. In addition, industries specializing in the construction of machines or structures of certain types have set up their own standards of allowable stresses to be applied to the design and construction of their products. In some cases, such as the building industry, these standards have been incorporated into the building laws and codes of municipalities to safeguard the public. Where no existing standards are available, it is up to the skill, judgment, and experience of the designer to establish the proper allowable stresses for any given set of conditions. To aid in this process, the data on mechanical properties of materials tabulated in MIL-HDBK-5A[6] are recommended for use in the design of military vehicles.

4-1.2.3 Safety Factors

4-1.2.3.1 General Discussion

The term *safety factor* or *factor of safety* has become widely used to denote the ratio of the design of the design load to the applied load. The design load was defined as the theoretical failure load used in the stress computations (see par. 4-1.2.2.2), and failure may be either the point of plastic deformation or the point of material rupture. Expressed as an equation

$$\textit{Safety Factor} = \frac{\textit{Design Load}}{\textit{Applied Load}}$$

The need for a safety factor to provide a limit between the maximum applied load and failure of the structure arises from the uncertainties as to the mechanical properties of the materials used, the uncertainties as to the maximum loads and load characteristics to be resisted, and the uncertainties in the methods used to calculate the stresses in the structural members. In deciding on the importance of these uncertainties and in selecting a safety factor many factors must be considered, such as the significance of a failure of a structure or part with respect to damage to the entire system, damage to other equipment, damage to surrounding property, and physical injury or loss of human life; the anticipated useful life of the structure; the extent to which deterioration from the operational environment is likely to occur and its effect upon the load resisting capabilities of the structure; and the degree of periodic equipment maintenance and inspections anticipated. Every effort should be made to reduce these uncertainties as much as possible so that structures can be designed with minimum safety factors.

Safety factors should not be placed indiscriminately upon both the load and upon the allowable stress level. In conjunction with this admonition, it should be noted that some of the considerations applicable to the selection of allowable working stresses for materials—such as nonuniformity of materials, inaccuracy of stress analysis, uncertainties of conditions of service, etc.—are also considerations in the selection of a factor of safety. Careful discrimination must be made not to make allowances for the same factors twice.

A common cause of overdesign is the application of the safety factor to both the load and to the stress. In order to avoid this abuse, select a value for the safety factor at the start of the stress analysis and apply it to the load (applied load) only. Do not use safety factors as multipliers for stress values[7].

4-1.2.3.2 Suggested Safety Factors[6,7]

The following safety factors are suggested to obtain design loads from applied loads in the design of military vehicles:

a. Structures in general	1.5
b. Handling loads	Per Specifications
c. Bearings	Per MIL-HDBK-5A
d. Joints	Per MIL-HDBK-5A

The following safety factors should be applied in addition to the factor for general structures given above:

a. Castings (plain) 2.0

b. Castings (with X-ray, close quality control and minimum elongation of 7 percent) 1.2

c. Fittings 1.2

d. Gun recoil loads that are a hazard to personnel 1.6

e. Gun travel lock when loss results in a hazard to personnel 1.5

4-1.2.4 Margin of Safety

The margin of safety (MS) is a ratio that compares the amount by which the ultimate strength of a member, as finally designed, exceeds the design load. The design load, as explained in par. 4-1.2.2.2, is the maximum probable applied load (par. 4-1.2.2.1) multiplied by a specified safety factor. The ultimate strength of the member is its theoretical calculated strength based upon the dimensions established for it by the final design; thus it is the actual final allowable load. Stated in the form of an equation:

$$MS = \frac{Final\ Allowable\ Load}{Design\ Load} - 1$$

In a well-designed structure, the margin of safety should be zero or have a slight positive value. The positive margin of safety is that overdesigned portion of a structure that cannot be practically eliminated. In the case of a redundant type of structure, where a failure of an individual element would result in the load being carried by other elements of the structure, a negative margin of safety may be acceptable.

4-1.3 STATIC LOADS

The evaluation of the static loads experienced by vehicle bodies and hulls generally presents no untoward difficulties to the designer. Static loads are the loads imposed upon the structure by the dead weights of the various assemblies and components that are mounted to it—such as the power plant, transmission, transfer case, final drives, fuel tanks (full), suspension components, fixed weapons, installed electronic equipment, turrets, booms, winches, etc.; the dead weight of the cargo to be carried; the weights of the crew and passengers; the weights of the various stowage items, such as ammunition, tools, special kits, repair parts, etc.; and the weights of the structural elements themselves. Thus, the evaluation of the static loads requires the determination of the weights of the items involved and the magnitudes and directions of the reaction-forces acting at the mounting points of each item. The latter is simply accomplished through the application of the basic principles of statics which state that, for equilibrium conditions to exist, the summation of forces along each of the three coordinate axes must be equal to zero and the summation of moments about each of these three axes must also be zero. In these calculations, the weights of the components are assumed to be concentrated at their centers of gravity; thus the locations of the centers of gravity of each component with respect to its mounting pads must also be known in order to evaluate the reaction forces at the mounts. The forces that act upon the hull structure are equal in magnitude but opposite in direction to the reaction forces on the component's mounting pads. In this manner, the static forces acting on every point of the structure can be determined and located. For the sake of expediency, however, only the forces due to the major components are considered.

The maximum loads applied to a military vehicle, however, are dynamic loads characterized by high shocks induced by rough terrain, by obstacles in the vehicle's path, and by the firing of heavy weapons. These forces are almost impossible to evaluate accurately; however empirical methods do exist that have resulted from long experience. Some of these empirical methods are discussed in par. 4-1.4. A knowledge of the static loads acting upon the various sections of the structure, however, is a basic requirement for a reasonable approximation of the dynamic loads.

4-1.4 DYNAMIC LOADS

Dynamic loads, as this term is used here, are impact loads or impact forces that result when the moving vehicle encounters rough terrain or obstacles in its path such as boulders, logs, walls, or ditches; when it plunges down after being driven rapidly over the brow of a narrow hill or a parapet; when it is struck by massive or rapidly moving objects; when it experiences the recoil of heavy weapons mounted upon it; or when it is subjected to the nearby blast of high explosives.

The procedures generally used to determine forces acting upon bodies and their effects are based upon the relationships between masses and accelerations, work and energy, and upon the assumption that the forces act upon rigid bodies during definite (comparatively large) intervals of time. Furthermore, when the forces vary in magnitude during the time period over which they act, the manner in which they vary is known, and this force-time history is incorporated into the mathematical procedures used. However, forces often act for extremely short (indefinite) periods during which neither their values at any instant nor their manner of variance (force-time history) are known. This is the nature of the dynamic loads experienced by the vehicle when it encounters the situations described earlier.

Loads or forces of this type are generally known as impulsive, or impact, loads. They produce extremely high pressures on the bodies upon which they act and result in appreciable deformations. This makes it incorrect to regard the bodies as rigid.

In most cases, the effects of impulsive forces on the motions of bodies are so great in comparison with the effects of other forces that may be acting that the effects of the other forces may be neglected during the impulse. The only details that can be readily determined of the change in the motion of a body that has received an impulsive load are its initial and final velocities. The distance traveled during the impact is indefinitely small; the time interval is also indefinitely small; hence the acceleration produced is indefinitely large, since the change in velocity is a finite amount. Thus, the distance, time, and acceleration factors are indeterminate. There is, however, a very definite and appreciable change in velocity; although, the manner in which the velocity changes during the interval of the impact is unknown, and only the initial and final velocities can be determined. Therefore, the momentum (mass times velocity) of the body at the beginning and end of the impact period are definite quantities. This permits problems that involve impact forces to be solved by applying the principles of impulse and momentum. Numerous references are available on this subject[8-12]. The principles are summarized in the following equations and demonstrated in a sample calculation.

$$\Sigma F_x \Delta t = M(v_{x_2} - v_{x_1}) \tag{4-1}$$

$$\Sigma T_o \Delta t = I_o(\omega_2 - \omega_1) \tag{4-2}$$

where

x = displacement in any direction, ft

F_x = force component in the given direction x, lb

Δt = duration of the impact, sec

M = mass of the body concerned, lb-sec^2/ft

v_{x_2}, v_{x_1} = final and initial velocities, respectively, of the body in the direction x, fps

T_o = turning moment about an axis O, ft-lb

I_o = mass moment of inertia of the body about axis O, ft-lb-sec^2

ω_2, ω_2 = final and initial angular velocities, respectively, of the body, rad/sec

The left-hand portion of Eq. 4-1 represents the linear impulse of the force system acting upon the body, and the right-hand portion represents the change in linear momentum experienced by the body. Similarly, in Eq. 4-2, the left-hand portion represents the angular impulse about an axis of rotation of the force system; and the right-hand portion represents the change in angular momentum of the body. These basic equations are often useful in evaluating impact forces that act upon military vehicles. The following illustrates the procedure applied to the impact experienced by the front wheel of a tracked vehicle when it encounters an abrupt bump in its path. Fig. 4-1 shows the forces acting on the road wheel at the instant of impact. When the principles of Eq. 4-1 are applied

$$-F_x \Delta t = M(v_{x_2} - v_{x_1} \tag{4-3}$$

The wheel has a velocity v_1 just before impact and v_2 the instant after. The magnitudes of v_1 and v_2 are the same; however, the direction of v_2 has changed during the impact. Therefore,

$$v_{x_1} = v_1$$

$$v_{x_2} = v_1 \cos\theta$$

$$\Delta t = \frac{\theta}{\omega_1} = \frac{R\theta}{v_1}$$

where

R = rolling radius of wheel

Substituting these expressions in Eq. 4-31 and solving for F_x result in the following expression

$$F_x = \frac{Mv_1^2 \ (\cos\theta - 1)}{R\theta} \quad (4\text{-}4)$$

When a similar procedure is applied to the forces and velocities acting in the y direction, the following results are obtained

$$\Sigma F_y \Delta t = M(v_{y_2} - v_{y_1}) \quad (4\text{-}5)$$

$$v_{y_1} = 0$$

$$v_{y_2} = v_1 \sin\theta$$

Substituting these values in Eq. 4-4 gives the following results

$$(F_y - P - W - ky)\,\Delta t = M\,(v_1 \sin - 0) \quad (4\text{-}6)$$

Substituting $\Delta t = \frac{R\theta}{v_1}$ and solving for F_y

$$F_y = \frac{Mv_1 \sin\theta}{R\theta} + P + W + ky \quad (4\text{-}7)$$

where

P = static load carried by the wheel
W = weight of the wheel assembly
ky = maximum downward force developed by the spring system as the wheel rises over the obstacle

These three forces, particularly the first two (P and W), are often omitted, because their contribution to the total impact force is minor in comparison to that of the first term. The ky term may or may not be significant depending upon the spring constant k. This is illustrated in the example which follows.

The total force F experienced by the wheel as it goes over the bump is the resultant of the two components F_x and F_y and can be calculated by means of the following equation

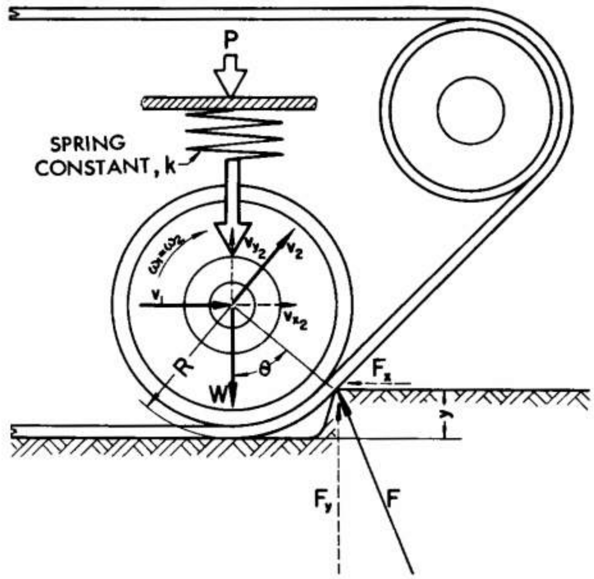


Figure 4-1. Forces on Road Wheel Striking an Obstacle

$$F = \sqrt{F_x^2 + F_y^2} \quad (4\text{-}8)$$

Sample Problem

Consider a 20-ton, tracked vehicle with 10, 24-in. diameter road wheels and a 1-in. high track that encounters an abrupt bump 5 in. high while traveling at 30 mph. Additional conditions of the problem are: total sprung weight of the vehicle is 32,000 lb and is carried equally by the 10 road wheels (3,200 lb per wheel); effective weight of the road wheel, tire, and road arm assembly is 700 lb; vertical spring constant at the wheel hub of the torsion bar suspension spring is 1800 lb per inch of deflection. Calculate the impact force experienced by a front road wheel of the vehicle.

If Fig. 4-1 is used to represent the front road wheel and the same notations are retained that are used in the figure, the following are known or can be readily determined.

P = 3,200 lb

W = 700 lb

$M = \frac{W}{g}$ = 21.7 lb-sec²/ft

I_o = 12.7 ft-lb-sec²

$R = 13$ in. (= 1.08 ft)

$k = 1{,}800$ lb/in. (= 21,600 lb/ft)

$y = 5$ in. (= 0.417 ft)

$v_1 = 30$ mph (-44 fps)

$$\theta = \text{arc cos}\,\frac{(R - y)}{R} = \text{arc cos}\,0.61538 = 52^\circ$$

$$= 0.9076 \text{ rad}$$

When the appropriate values are substituted in Eqs. 4-4, 4-7, and 4-8 and the equations are solved, the following results are obtained

$$F_x = \frac{21.7 \times 44^2\,(0.61538 - 1)}{1.08 \times 0.9076} \tag{4-9}$$

$$= 16{,}485 \text{ lb}$$

$$\begin{aligned} F_y &= \frac{21.7 \times 44^2 \times 0.78823}{1.08 \times 0.9076} + 3{,}200 + 700 \\ &\quad + 1{,}800 \times 5 \\ &= 33{,}114 + 3{,}200 + 700 + 9{,}000 \\ &= 46{,}014 \text{ lb} \end{aligned} \tag{4-10}$$

$$F = \sqrt{(-16{,}485)^2 + (46{,}014)^2} \tag{4-11}$$

$$= 48{,}878 \text{ lb}$$

Eq. 4-10 illustrates the relative importance of the four terms. In this case, the magnitude of the last term is appreciable while that of the third term is negligible. The designer must exercise engineering judgment in deciding whether or not to omit certain terms.

4-1.4.1 Road Loads

Many problem situations involving dynamic loads can be solved by means of the equations and procedures given in par. 4-1.4. Unfortunately, most practical situations cannot be simplified to resemble anything comparable to the conditions used in the preceding sample problem. Vehicles that travel over rough terrain in real situations, particularly military vehicles, are in a constant state of random pitching, rolling, and yawing during which they experience violent shocks from obstacles in their paths such as holes, ruts, ditches, rocks, logs, and walls. These shocks may be initiated at any portion of the vehicle that contacts the terrain (obstacles are here considered as part of the terrain) and may be in any direction, i.e., horizontal, vertical, or lateral. This results in an almost hopelessly complex dynamic system that is practically impossible to evaluate by means of rigorous mathematical procedures. Since these loads arise from the characteristics of the terrain over which the vehicle is traveling, they are referred to as road loads to distinguish them from other types of loads such as weapon recoil loads, ballistic impact loads, blast loads, etc.

At the present time, the determination of the basic road loads to which a vehicle may be subjected involves the application of a semi-empirical procedure that has been developed at the U. S. Army Tank-Automotive Command[7]. It is the result of experimental procedures based upon data collected from actual vehicle tests involving such indicators as oil pressure readings, strain gage and accelerometer measurements taken while the vehicles were in motion, from the study of broken parts and the application of stress calculations to determine the loads that produced failure, and from the study of the performance of similar vehicles under tactical or simulated tactical conditions. It is not a rigorous method, nor is it to be considered indisputably accurate. The procedure is extremely simple but has been found to yield reasonably accurate results in approximating the ultimate forces that act upon different vehicle types during typical tactical operations. It is believed to be the best system currently available for this purpose and one that can be easily modified either upward or downward should more complete experience so indicate. Before proceeding with an explanation of this system, it should be realized that the load values thus determined are the maximum design loads or loads at which failure of parts is expected. Refer to par. 4-1.2.2.2 for a discussion of design loads. Maximum applied loads are generally taken as 65 to 70 percent of the design loads.

The procedure for estimating maximum road loads involves the determination of a basic load factor n for the vehicle type, weight, and class of service being considered and then using this

factor as a multiplier in several simple equations that are given later. Fig. 4-2 is a simple nomograph for determining the basic load factor *n*. It shows seven vehicle classifications (represented by the seven sloping lines) rated according to vehicle types (trailers, wheeled vehicles, tracked vehicles) and in accordance with their intended missions or operating environments (logistic vehicles, combat vehicles, vehicles limited to paved roads, etc.). The horizontal scale represents the gross vehicle weights in thousands of pounds and the vertical scale represents the basic load factors n.

The sloping lines on the nomograph are a

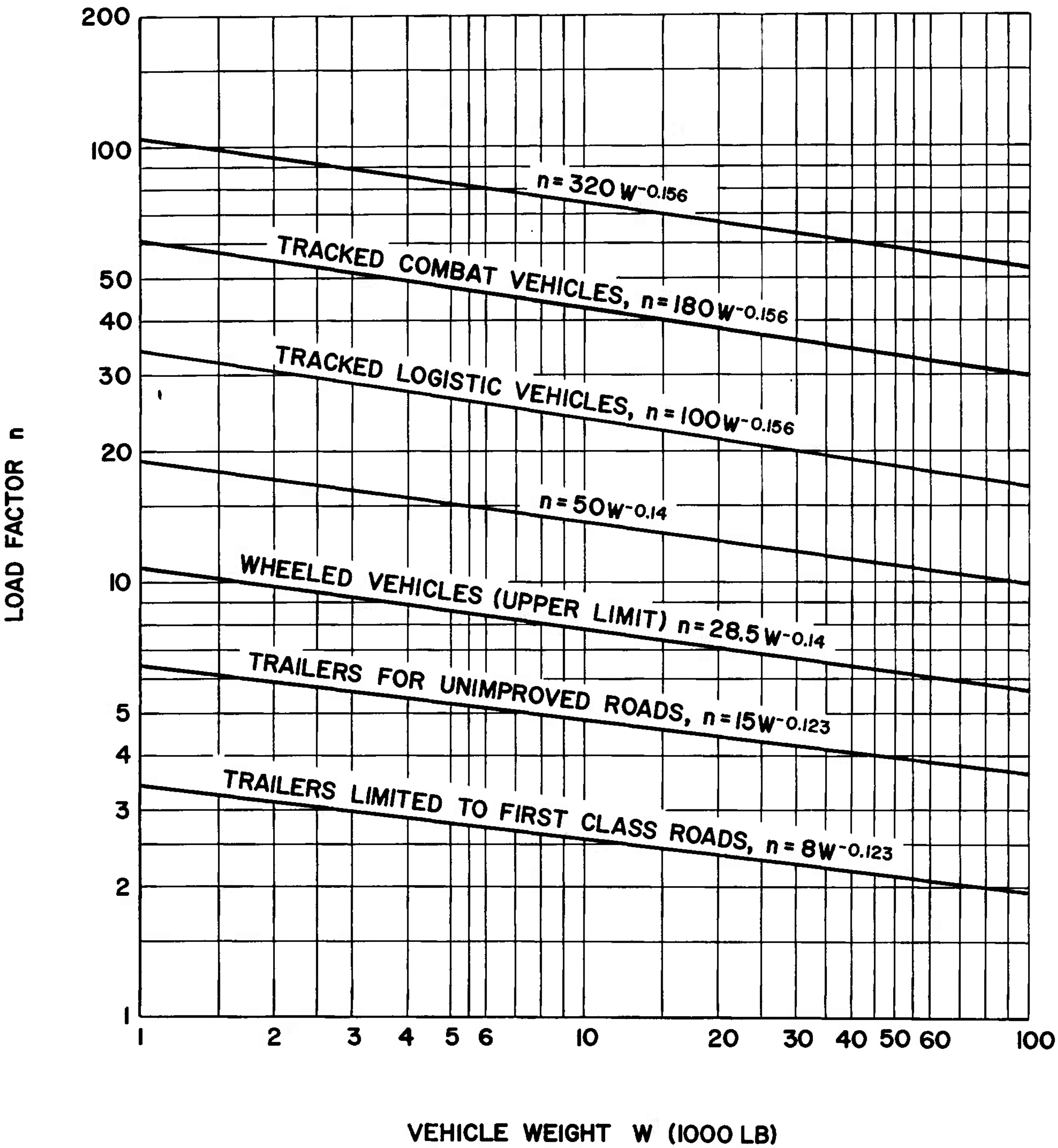


Figure 4-2. Load Factor n for Estimating Road Loads

log-log plot of the general form $n = CW^k$ where the exponent k is the slope of the line and the constant C locates the line in the vertical direction. It is also the n intercept, or the value of n when weight equals one pound. The numeric values assigned to C and k for the seven vehicle classifications shown seem reasonably correct in the light of present data and experience. Should future experience indicate otherwise, the lines can be repositioned to suit the new data by selecting appropriate values for C and k. Increasing or decreasing C while maintaining k constant raises or lowers the line while maintaining the slope of the original line. Increasing or decreasing the absolute value of k increases or decreases, respectively, the slope of the line about its intercept. It should be noted, however, that the intercept is at W=1 lb and, therefore, does not appear in Fig. 4-2 which ends at W=1,000 lb.

For typical track-laying vehicles that have rigid hulls—such as tanks, self-propelled artillery, and armored carriers—the load factor n determined from Fig. 4-2 represents the maximum acceleration in g's that may be experienced by the front end of the vehicle during severe tactical operations. At successive stations along the vehicle length, the load factor decreases at a uniform rate (straight line function) from a maximum at the front to a mimimum at the vehicle center of gravity n_{CG} and is then considered to remain constant for subsequent stations from the vehicle center of gravity to the rear end of the vehicle. The value of n_{CG} is taken as $\frac{n}{2}$ or as $n(1 - \frac{\ell}{L})$, whichever gives the greater value, where ℓ is the horizontal distance from the front of the vehicle chassis to the center of gravity and L is the overall length of the chassis.

When applied to trailers, the load factor n is used as a general load factor that has a constant value at all stations along the trailer length.

The application of this system of design load estimating to wheeled vehicles has been very limited thus far. It seems reasonable at this time, however, to use the load factor n as a general load factor for the vehicle as is done in the case of trailers. Engineering judgment may be exercised in special cases where the wheeled vehicle design configuration and operational characteristics approach those of rigid-hulled tracked vehicles.

To use the nomograph of Fig. 4-2, first determine the vehicle classification and operational category and select the appropriate curve on the figure. The severity of the categories increase as we go up on the chart. Since the classification curves are approximately parallel lines, intermediate categories can be created between the existing curves to suit special conditions. For example, if the vehicle is to be a tracked logistic type but, in the judgment of the design engineer, will be subjected to greater abuse than is normal for a vehicle of that type and yet perhaps not quite as severe as that experienced by combat vehicles (which would be the next standard category), a special classification line can be drawn between the lines that represent these two types of vehicles. This is the significance of the two unlabeled curves on the chart. Having selected an appropriate classification curve, determine the load factor n at the intersection of the selected curve with the estimated gross weight W of the vehicle.

The design load on the sprockets and road wheels of typical track-laying vehicles can be estimated by applying the equations which follow. It should be understood, however, that these loads do not occur simultaneously but are the maximum values that may occur under dynamic conditions at the location indicated.

a. Horizontal and vertical design loads on forward sprocket, considered to act at centerline of track and directed rearward and upward, respectively,
$= 0.15nW$, lb

b. Vertical load on No. 1 road wheel, considered to act upward on centerline of road wheel,
$= 0.1nW$, lb

c. Vertical load on No. 2 and all successive roadwheels,
$= 0.05nW$, lb

d. Vertical and horizontal loads on rear sprocket, considered to act upward and forward, respectively,
$= 0.07nW$, lb

e. Transverse loads acting horizontally against bottom edge of road wheel and directed inward

toward vehicle center, load per road wheel

$$= \frac{W}{2}, \text{ lb}$$

Sample Problem

Consider a tracked combat vehicle that is expected to have five road wheels per side and a gross weight of 20 tons. From Fig. 4-2, $n = 34$

Vertical and horizontal loads on forward sprocket

$= 0.15 \times 34 \times 40{,}000 = 204{,}000$ lb

Vertical load on No. 1 road wheel

$= 0.1 \times 34 \times 40{,}000 = 136{,}000$ lb

Vertical loads on No. 2, 3, 4, and 5 road wheels

$= 0.05 \times 34 \times 40{,}000 = 68{,}000$ lb

Vertical and horizontal loads on rear sprocket

$= 0.07 \times 34 \times 40{,}000 = 94{,}200$ lb

Transverse load on bottom edge of each road wheel

$$= \frac{40{,}000}{2} = 20{,}000 \text{ lb}$$

The discussion of par. 4-1.4.1 thus far pertains to the estimation of design loads experienced by major components of the vehicle, such as the hull, turret, and suspension system components, but does not apply to shock loads experienced by small items mounted in, or on, the vehicle. A small item is here defined as any item whose weight is less than 3 percent of the weight of the vehicle. Data developed from studies of such small parts that failed in service due to shock loading reveal that the accelerations experienced by these parts are more severe than the maximum acceleration experienced by the vehicle at the mounting location of the part concerned. This may seem incredible at first but gains credibility after a little reflection.

The shock experienced by the vehicle is transmitted through the hull and structural members to the various components in contact with the vehicle structure and causes these elements to vibrate at, or very near, their natural frequencies. The accelerations inherent in these vibrations are often far more severe than that of the shock that initiated the vibration. This can be illustrated by means of the following example.

Consider a wall of an armored vehicle hull that has received a 20-g impact which caused the wall to vibrate at its natural frequency of 100 cps and with an initial amplitude of 0.2 in. The following equations describe the motion of a freely oscillating body having one degree of freedom:

$$y = A \sin \omega t \tag{4-12}$$

$$\dot{y} = \omega A \cos \omega t \tag{4-13}$$

$$\ddot{y} = \omega^2 A \sin \omega t \tag{4-14}$$

where

y = amplitude of vibration, in.

$\dot{y}$ = velocity, in./sec

$\ddot{y}$ = acceleration, in./sec

A = maximum amplitude, in.

ω = angular velocity, rad/sec

t = time, sec

It is evident from Eq. 4-14 that the acceleration reaches its maximum value when $\sin \omega t = 1$, in which case

$$\ddot{y}_{(max)} = -\omega^2 A \tag{4-15}$$

The minus sign merely indicates the direction of the acceleration. By substituting numerical values from our hypothetical vehicle into Eq. 4-15, we can get a fairly good approximation of the magnitude of the maximum acceleration.

$$\ddot{y}_{(max)} = (2\pi \cdot 100)^2 \times 0.2$$

$$= 7.9 \times 10^4 \text{ in./sec}^2$$

$$= 204 \text{ g}$$

This is a simplified treatment and merely serves to illustrate a point. In the actual case, the hull wall is not a freely vibrating body nor does it vibrate in a purely sinusoidal manner. Its vibration pattern is a very random conglomerate of many vibrations from a variety of sources. This makes it virtually impossible to predict by analytical methods the maximum acceleration that it may experience. However, the example does serve to illustrate how it is possible for components mounted to a vehicle hull to experience accelerations resulting from external shocks to the vehicle that are greater than those associated with the original shock.

Since useful analytical method for estimating maximum accelerations experienced by small components mounted to a vehicle hull are not available, a semi-empirical procedure similar to the one given for estimating basic road loads was also developed at the U. S. Army Tank-Automotive Command[7]. This requires the construction of a shock factors diagram similar to the one shown in Fig. 4-3. A separate diagram is required for every different vehicle type and weight. The procedure follows.

From the nomograph (Fig. 4-2) and the discussion accompanying it, determine values for n, n_{CG}, and C for the vehicle type, weight, and service classification being considered. On a suitable log-log grid, select a horizontal scale that will accommodate small item weights ranging from 1 lb to 3 percent of the vehicle weight under consideration. Similarly, select a vertical scale that will accommodate shock factors ranging from the value determined for n_{CG} to that determined for C. On a vertical line representing 3 percent of the vehicle weight, locate points corresponding to the values of n and n_{CG}. Locate the value of C on the vertical axis, and connect it with the two points n and n_{CG} (see Fig. 4-3). Extend the two lines horizontally from the 3 percent vehicle-weight-line to the right-hand limit of the chart at n and n_{CG}. This results in two curves. The upper yields the maximum shock factors, in g's, that may be experienced by small items of various weights when mounted at the front end of the vehicle; and the lower curve yields a comparable factor for small items mounted at the vehicle center of gravity. Shock factors for items mounted between the front end and the center of gravity can be extrapolated from these two limits. The shock factor is considered to remain constant at stations aft of the center of gravity.

Fig. 4-3 is an example of a shock factors diagram for a tracked combat vehicle weighing approximately 15,500 lb, with an overall chassis length L of 120 in., and its center of gravity located 75 in. from its front end ($\ell = 75$ in.) Fig. 4-2 indicates that this type, class, and weight of vehicle will have a basic load factor n of 40 g and a value for C of 180 g. Since $\frac{n}{2}$ is greater than $n(1 - \frac{\ell}{L})$ in this case, n will equal 20 g. The limitation of 3 percent of the vehicle weight in the definition of "small items" results in a maximum weight for these items of 465 lb. These values can be easily identified in Fig. 4-3. Thus, according to Fig. 4-3, a 50-lb item located about midway between the front end of the vehicle and the center of gravity may experience a maximum shock of approximately 55 g.

4-1.4.2 Weapon Recoil Loads

When weapons are fired from a vehicle, the recoil forces are transmitted through the weapon mounting to the vehicle structure and from there through the suspension system and spades (if present) to the ground. Reactions from the earth may be distributed uniformly over the spade area, or they may be concentrated due to large rocks. Investigations should be conducted of the effects of recoil loads for different angles of elevation and azimuth, based on the assumption that there is ledge rock in the earth beneath the vehicle. The most serious recoil loads are produced by the large caliber guns that constitute the main armament of self-propelled artillery, tanks, and vehicle-mounted antiaircraft artillery; but weapons of smaller caliber, particularly rapid firing weapons, also merit serious consideration. These latter types of weapons subject the vehicle structure to rapidly repeated shock loads which lead to failure due to fatiguing of materials.

Loads resulting from the recoil of weapons are probably the most accurately known of all loads acting upon the military vehicle. They are calculated by the application of established methods of dynamics and are confirmed by test firings. The procedure is based upon the principle of the conservation of momentum. The momentum (mass times velocity) of the projectile M_p as it leaves the muzzle plus the momentum of the propelling gases and unburned powder M_g are equated to the momentum of the recoiling parts M_r as they move in a direction opposite that of the projectile, or

$$M_p V_p + M_g V_g = M_r V_r \qquad (4\text{-}16)$$

where V represents velocity, and the subscripts p, g, and r signify the projectile, gases, and recoiling parts, respectively. The velocity of the recoiling parts can be determined as

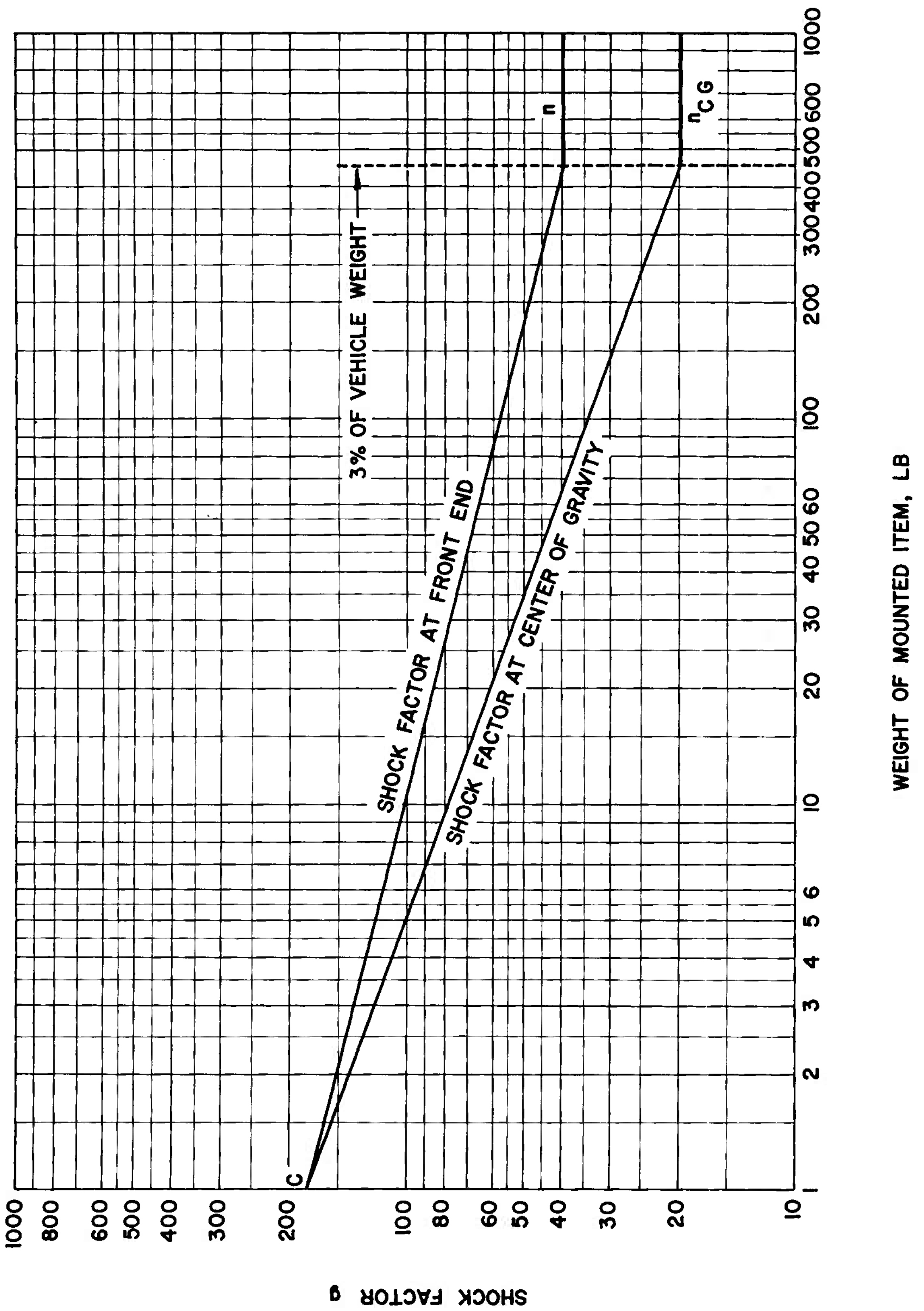


Figure 4-3. Shock Factors for Small Items Mounted in the Vehicle

$$V_r = \frac{M_p V_p + M_g V_g}{M_r} \quad (4\text{-}17)$$

Once the velocity of the recoiling parts is determined, and knowing the length of the recoil stroke L_r, the duration of the recoil t_r can be calculated as

$$t_r = \frac{L_r}{V_r} \quad (4\text{-}18)$$

Most recoil mechanisms are designed to give a constant resistive force throughout their working stroke. This force can be readily determined by equating the momentum of the recoiling parts to the impulse (force times time) it produces on the trunnion mounting.

$$M_r V_r = F_r t_r \quad (4\text{-}19)$$

from which

$$F_r = \frac{M_r V_r}{t_r} \quad (4\text{-}20)$$

A more detailed procedure for determining weapon recoil forces is given in Ref. 13. Recoil forces developed by several standard heavy armament systems are shown in Table 4-2. Recoil forces developed by cal. 50 and 20-mm weapons can be found in Ref. 14. Under some conditions, short duration peaks occur in the recoil cylinder oil pressure which indicate greater recoil loads than those shown in Table 4-2. These peaks usually have a duration of less than 3 msec and have no appreciable effect upon the vehicle structure.

4-1.4.3 Blast and Ballistic Impact

4-1.4.3.1 Blast

High-energy blast is a loading condition more common to military vehicles than to civilian vehicles. The most severe blasts are those produced by the detonation of nuclear weapons. Blast, or more properly air blast, is a pressure wave that emanates from the detonation of a high-energy explosive. It is characterized by an abrupt rise in ambient pressure above the normal barometer (overpressure) followed by a gradual decay to a zero overpressure. The magnitude of the overpressure from a nuclear burst depends upon many factors, the most significant being the yield of the weapon used, the height of the burst above the ground surface, and the distance to ground zero. Near surface bursts with yields in the megaton range produce overpressure in excess of 100 psi at a distance of 1000 yd from ground zero. The same weapon detonated at an altitude of 6,500 ft will produce an overpressure of about 16 psi at a distance of 1.3 miles from ground zero, so that the total air pressure will be more than double the normal atmospheric pressure.

Strong transient winds are associated with the passage of the pressure front. These blast winds are considerably stronger than the ground winds (or afterwinds) that are due to the updraft caused by the rising fireball and which occur at a later time. The blast winds may have peak velocities of several hundred miles per hour fairly close to ground zero. Even at more than six miles from the explosion of a 1-megaton nuclear weapon, the peak velocity will exceed 70 mph. Such strong winds can contribute greatly to the blast damage caused by nuclear weapons, particularly to vehicles which can be tumbled by the strong winds and, thus, damaged as they strike the ground.

When the front of the shock wave strikes the face of a structure, reflection occurs. As a result of this reflection, the overpressure builds up rapidly to at least twice (and generally to several times) that in the incident wave front. The actual overpressure attained is a function of such factors as the value of the peak overpressure of the incident shock wave and the angle that the direction of propagation of the shock wave makes with the face of the structure. As the wave front moves forward, the reflected overpressure on the face of the structure drops rapidly until the total overpressure is that of the shock wave without reflection, but added to it is the dynamic pressure of the blast wind. At the same time, the shock wave bends, or, more properly, diffracts around the structure, so that the structure is eventually engulfed by the shock. At this time, approximately the same pressure (than of the incident shock wave) is experienced by the side walls and roof and, finally, by the rear wall. The front wall, however, is still subjected to the dynamic wind pressure, although, the back wall is shielded from it. A detailed treatment of this

TABLE 4-2 NORMAL RECOIL LOADS OF VARIOUS VEHICLE-MOUNTED ARMAMENT SYSTEMS

Vehicle	*Mount*	*Armament*	*Average Recoil*	
			Force, lb	*Length,* in.
Tanks				
M41 Series	M76 Series	76 mm Gun, M32	28,400	10
M47	M78	90 mm Gun, M36	55,800	12
M48 Series	M87 Series	90 mm Gun, M41	87,000	12
M60 Series	M116 Series	105 mm Gun, M68	94,000	12
M103 Series	M48	120 mm Gun, M89	120,000	13
Self-propelled Artillery				
M56	M88	90 mm Gun, M54	17,800	36-43
M52A1	M85	105 mm How., M49	59,300	12
M108	M139	105 mm How., M103	52,300	12
M44	M80	155 mm How., M45	133,000	19
M53	M86	155 mm Gun, M46	93,200	18
M109	M127	155 mm How., M126	132,800	23
			84,900	36
M107	M58	175 mm Gun, M113	188,500	31
			88,000	72
M55	M86	8 in. How., M47	234,600	18
M110	M158	8 in. How., M2A1E1	188,000	31
			65,000	72

subject—both shock waves and dynamic wind—including methods of evaluating the magnitudes of the pressures involved from various yield weapons and at various distances is given in Ref. 15.

The forces that act upon a vehicle body or hull structure as a result of air blast from nuclear weapons are generally very large in comparison with similar loads encountered in conventional engineering. In addition, they are transient in character and their probability of occurrence is small, although this latter statement may be subject to re-evaluation in the future. For an economical design concept under these conditions, the full resistance potential of the structure should be utilized and not limited by any weak links in the system. Spherical and cylindrical arch-shaped structures are the most efficient in this respect because of their inherent resistance to external pressures. Structures with sloping surfaces tend to reduce the reflected overpressure. Joints and attachment points, including fasteners used, should be evaluated in detail. All modes of failure, i.e., buckling, shear, flexure, bond, etc., should be considered. Large flat unsupported areas should be avoided. It is desirable to avoid brittle materials and brittle types of failure wherever possible. Plastic deformation of structural elements may be permissible at the design loads in many applications (see par. 4-1.2.2.2). Factors of safety in the conventional sense (par. 4-1.2.3) are generally not used in the design process involving nuclear blast loads. The design loads involving overpressures and dynamic wind loads due to blast can only be based upon estimates, and increased factors of safety may result in excessively overdesigned structures.

During dynamic loading conditions of the type associated with high-energy blast, materials exhibit higher yield strengths than they do under static loading conditions. Every advantage should be taken of this property so as not to penalize the design unnecessarily. Toward this end, the yield point strengths given in Table 4-3 should be used in determining the plastic strengths of members rather than the conventional values of yield point or standard specified minimum yield stresses that are customarily used. The procedures involved in analyzing the loads induced in variously shaped structures by shock and dynamic loading, and in

TABLE 4-3 RECOMMENDED YIELD STRESSES FOR STEEL WHEN SUBJECTED TO DYNAMIC LOADS

Application	*Tension or Compression*, lb/in.^{2}	*Shear*, lb/in.^{2}	*Bearing* lb/in.^{2}
Structural Members, Carbon Stl. (ASTM-A7)	42,000	25,000	
Welds	42,000	25,000	
Rivets (ASTM A-141)	40,000	30,000	60,000* 80,000†
Rivets, High Strength, (ASTM A-195)	60,000	40,000	80,000
Bolts, High Strength, (ASTM A-307)	50,000	30,000	60,000
High Strength Alloy Steels	1.1 times min static yield strength	0.6 times tensile yield strength	

*single shear †double shear

determining the sizes and arrangements of members to withstand such loads are too involved for this handbook. A very practical treatment of this subject is given in Refs. 15 and 16.

Thus far, this discussion of blast loading has pertained to shock waves produced by extremely large detonations that engulf the entire vehicle. Another type of blast loading experienced by military vehicles is a localized blast affecting only a portion of the vehicle. Such blasts are produced by land mines exploding under the vehicle, grenades and various demolition charges detonated in close proximity to the vehicle, explosive projectiles which detonate as they strike, and by near misses of aerial bombs. Although the total energy released in these types of blasts is much less, by many orders of magnitude, than in the nuclear detonations, it is sufficient to produce local damage to vehicle structures and may even put the vehicle completely out of action. This is possible because the energy is released in such close proximity to the vehicle that the shock is not sufficiently attenuated by distance.

The shock loads produced by these localized high-energy blasts are very difficult to evaluate. Orthodox military explosives develop a shock energy of approximately 700 ft/ton/lb of explosive and a maximum pressure at the surface of the charge of from 100 to 200 kbar, depending upon the density and particular type of explosive used (TNT develops about 150 kbar). This energy is released in all directions to

form a pressure wave that travels at transonic velocity. Since this pressure wave moves outward in all directions from the charge, not all of the energy released is effective against the vehicle. Furthermore, as the pressure wave moves through the atmosphere, it is attenuated very rapidly with distance according to the following approximate relationship

$$P = 1500\left(\frac{W^{1/3}}{R}\right)^{2.25} \quad (4\text{-}21)$$

where

P = peak pressure, psi
W = weight of explosive charge, lb
R = distance to charge, ft

Eq. 4-21 is based upon the effects of TNT. For other explosives, the factor W should be adjusted in accordance with the relative strength of the explosive used. Additional factors—such as charge shape, charge density, detonating techniques used, strength characteristics of the casing that contains the explosive and the presence of pressure reflecting surfaces (such as the ground)—all influence the effect of the charge. Analytical procedure for accurately quantifying the effects of localized high-energy blasts on military vehicles is extremely complex. The best procedure at the present time is to base design calculations upon data obtained from field tests of material exposed to high-energy blasts identical to those under consideration. Guidance of this type is given in Ref. 17.

4-1.4.3.2 Ballistic Impact

Ballistic impacts relate to the impacts produced by projectiles and large fragments that strike the vehicle. The severity of such impacts depends upon the mass and velocity of the missile at the time of impact, upon the area of the contact, the angle of incidence the flight path makes with the impacted surface, the condition of the struck surface, and upon the resistance offered by the surface struck. These are discussed in greater detail in par. 2-13.

The average force developed during the impact of a missile against the surface of a vehicle or a vehicle component can be predicted with reasonable accuracy by applying the principles of impulse and momentum that were discussed in par. 4-1.4. The impulse $F\Delta t$ is equated to the change of momentum experienced by the projectile as a result of the impact, and the average force F can be calculated by dividing the impact by the time interval Δt, which can be measured, calculated, or assumed.

Example

Consider the impact of a 37 mm, 1.9-lb projectile traveling in a horizontal direction that strikes the front of a tank body with a velocity of 2,200 fps and ricochets downward at an angle of 60° to its original path and with a departing veolocity of 900 fps. Using the same notation as was used in par. 4-1.4 and letting the subscripts x and y denote horizontal and vertical directions, respectively, we can write the following

$$M = \frac{1.9}{32.2} = 0.059 \text{ lb-sec}^2/\text{ft}$$

$$v_1 = v_{x_1} = 2{,}200 \text{ fps}$$

$$v_2 = 900 \text{ fps}$$

$$v_{x_2} = -900 \cos 60^\circ = -450 \text{ fps}$$

$$v_{y_1} = 0$$

$$v_{y_2} = -900 \sin 60^\circ = -780 \text{ fps}$$

$$F_x\Delta t = 0.059(-450 - 2{,}200) = 0.059(-2650)$$

$$= -156.35 \text{ lb-sec}$$

$$F_y\Delta t = 0.059(-780 - 0) = -46.02 \text{ lb-sec}$$

$$F\Delta t = \pm\sqrt{(-156.35)^2 + (-46.02)^2}$$

$$= \pm\sqrt{26563} = -162.98 \text{ lb-sec}$$

and assuming $\Delta t = 5$ msec

$$F = \frac{-162.98}{0.005} = -32{,}596 \text{ lb}$$

The negative sign merely indicates that the reaction force of the vehicle body on the projectile is opposite to the projectile velocity.

4-1.4.4 Cargo-imposed Loads

Cargo-imposed loads affect the vehicle in several ways. The most obvious is the actual dead weight of the cargo upon the cargo floor. This requires normal design techniques to distribute this load to the structural members and thence, through the vehicle suspension system, to the ground. However, consideration should be given to the effects of vehicular accelerations upon these cargo-imposed loads. Upward accelerations experienced by the cargo bed due to road irregularites multiply the effects of the load upon the floor. Similarly, sudden downward accelerations increase the forces experienced by cargo tiedown devices. The same applies to the effects of lateral and longitudinal accelerations. Load requirements of tiedown devices are given in par. 3-41. Anchors for tiedown devices on floors, walls, etc., must have comparable strengths in order to be effective.

Consideration should also be given to the effects of bulk loaded cargo. Since bulk loaded cargo is not restrained by tiedowns, under certain conditions of vertical displacement the cargo may leave the floor when the latter starts downward only to be met during its descent by the again rising floor. The resulting impact loading may exceed 2½ g.

4-1.4.5 Auxiliary Equipment Loads

Auxiliary equipment—such as cranes, winches, dozer blades, hoists, etc.—bring forces to bear upon the vehicle structure that differ from the road loads and should be considered separately. This type of equipment subjects the vehicle to overturning moments which produce bending and twisting in structural members. Vehicle frames and bodies must generally be stiffened considerably to accommodate these types of auxiliary equipment. The exception to this is the heavily armored type of hull which generally has an excess of stiffness that is inherent in that type of design.

When making a force analysis of a vehicle equipped with auxiliary devices of the type mentioned, due consideration must be given to all angles and modes in which the equipment may be used to determine the most severe conditions. A crane lifting over the side of a vehicle has a greater tendency to overturn the vehicle than when lifting over the ends. Outriggers or stabilizers may be required to permit this type of operation and the vehicle frame must be stiffened accordingly. Lower load limits are often specified for side lifts than for lifts made over the end of the vehicle.

Special considerations must also be given to the attachment points of auxiliary equipment. These are the areas where the forces are concentrated, and the design should strive to distribute these forces to the supporting structure in a manner that will keep contact stresses as low as is practical.

4-1.4.6 Handling Loads

Handling loads such as are experienced when the vehicle is towed, hoisted, or transported on board other conveyances generally do not present any serious problems unless they are completely overlooked during the original design and provisions must be made for them later. Thus, the designer should realize at the beginning of the design phase that military vehicles are often required to tow other vehicles; to themselves be towed (from front or rear); to be hoisted aboard ships, trains, and when being repaired; and to be immobilized by lashings during shipments and during airborne and airdrop operations. The requirements at these various attachment points are discussed in pars. 3-40 and 3-41.

Design load requirements imposed on the vehicle by various transportation systems are not well defined in military publications. The following guidance, however, may be useful in this respect

a. Design all small items that may be stepped on to withstand a load of 300 lb.

b. For transportation by rail, design equipment to survive vertical accelerations of ±2 g, longitudinal accelerations of ±12 g, and transverse accelerations of ±3 g.

c. For transportation by ship, a load factor of 2 g in all directions has been found adequate.

d. For transportation by aircraft, consult Refs. 18 and 19.

4-1.4.7 Loads Encountered During Airdrop Operations

Shock loads encountered by vehicles during airdrop operations occur at several different times during the deployment of the parachute

system and upon contact with the ground. Designing for this specialized type of operation is thoroughly discussed in Refs. 19, 20, and 21. Ref. 19 is excellent on this subject. For general guidance, the impact experienced on ground contact by a vehicle that has been prepared for airdrop with a suitable shock absorbing system is about 15 g in the vertical direction. Without a suitable shock absorbing system, the shock experienced may be from 40 to 100 g.

4-1.4.8 Loads Associated With Amphibious Operations[165]

A discussion of some of the fundamental subject areas to be considered when designing for amphibious operations is given in pars. 4-5.3, 4-7.4, and 4-18 through 4-24. As pertains to loads associated with amphibious operations, it can be said that they are generally far less severe than those experienced in cross-country, off-road operations; so that the vehicle structure is generally inherently strong enough to withstand the loads encountered in water operations. Certain considerations may be significant in special cases, however, particularly where lightweight, thin-skinned vehicles are concerned. These bear pointing out lest they be overlooked.

One fundamental difference between operating as a land vehicle and operating as a water vehicle is the role played by the vehicle body. When operating as a land vehicle, loads are carried primarily by the frame members and the suspension system, while the body merely serves to contain the cargo, protect against weather, etc. When operating as a water vehicle, however, the suspension system has very little to do with transmitting the loads—in fact, it becomes little more than an unavoidable burden on the body. The body, or more properly the hull, now serves as the primary load-carrying element in addition to the functions it normally provides in a land vehicle. Buoyancy forces, hydrostatic forces, act upon the entire wetted surface of the hull; and for this reason, the hull plates must have sufficient strength and stiffness to resist these forces.

The pressure on the hull increases directly with the depth to which that point on the hull is submerged, and is equal to the product of the depth h and the weight per unit volume of water w; or, if p is the pressure per unit area, $p = hw$. The weight of a unit volume of fresh water is about 62.4 lb/ft^3, or 0.0316 lb/in^3. For sea water these values are 64 lb/ft^3 and 0.037 lb/in^3. Thus, the hydrostatic pressure experienced by the bottom plates of a hull that has a draft of 6½ ft (78 in.) in fresh water is $78 \times 0.036 = 2.81$ psi. This, by itself, may not seem impressive, but consider a rather insignificant 12×20 in. access plate on the bottom of the hull. Since this plate has an area of 240 in.^{2}, the total hydrostatic force acting on this plate is 674 lb.

Since the amphibious vehicle is fairly uniformly supported by the hydrostatic pressures on the hull, bending and twisting of the structure is not nearly as severe as it is in land operations. This is not true in the case of large ships, but it must be remembered that the military amphibious vehicle differs from large ships in two important respects— (1) it is relatively short with respect to the length of waves encountered, and (2) military vehicles—as distinguished from amphibians—are generally intended to operate on relatively calm inland waterways. Of these two, the first is the most significant. Since the length of the vehicle is such that it cannot span more than one wave, it is more-or-less free to rise and fall with each wave rather than be bent and twisted, first one way and then another, under the influence of two or more waves simultaneously as is the case with large ships.

4-1.4.9 Wind and Snow Loads

Wind and snow loads do not present any particular problem in the design of military vehicles except perhaps in the case of extremely thin-skinned vehicles, canvas-covered vehicles, or vehicles designed for unusual environments such as on high mountain tops or in the polar regions. The general practice in structural engineering is to assume a wind pressure of 15 psf on the walls of structures less than 60 ft above the ground and 20 psf on higher surfaces. On the projected areas of exposed structural-steel frames, the practice is to provide for 50 percent greater wind pressures than on vertical walls. Wind pressures normal to sloping roofs that are steeper than 4 in. vertical rise in 1 ft horizontal are taken as 1.5 psf for each inch vertical rise per 1 ft horizontal to a maximum of 20 psf. For extremely severe situations, or where wind speeds over 70 mph are expected, these values

should be increased. For a more detailed discussion of wind pressures on structures, particularly where more rigorous methods are desired, see Ref. 22.

Snow loads are merely a matter of weight resting on the roofs or other horizontal surfaces. Where vehicles are completely buried by extreme snowfalls or high drifts, vertical surfaces may be treated the same as when subjected to hydrostatic pressures. For these calculations, the specific gravity of the snow must be known or assumed. Unfortunately, there are many different types of snow, each with its own specific gravity, and the specific gravity of each type varies with its moisture content, temperature, age, and other factors. Thus, the following data (measured *in situ*) may be useful: the specific gravity of new snow varies from 0.03 to 0.15, of settling snow it is about 0.2, of settled snow it is about 0.4, of old snow it is about 0.5, and of spring snow and slush it is about 0.7. More detailed data on snow types can be found in Refs. 109 and 166.

4-2 WEIGHT, STRENGTH, AND STIFFNESS

4-2.1 MATERIAL SELECTION

The materials from which bodies and hulls are manufactured are selected on a number of criteria of unequal importance. These include the durability of the material, its strength-to-weight ratio, the cost of the material, and the cost of manufacture. A considerable amount of information on general material selection is presented in par. 2-8, therefore, only some specific considerations, mainly for metals, are presented here. Design information on structural plastics is contained in Refs. 23, 24, and 167.

In the design of military vehicles, it is important to provide the greatest strength for the least weight. Thus, we must consider minimum weight design. The mechanical properties most significant in the selection of materials are the moduli of elasticity E (slightly different in tension and compression for some materials) and rigidity G; the proportional limit, yield and ultimate stresses in tension, compression, and shear; and the density of the material. Additional considerations of ductility, toughness, resilience, and temperature, fatigue, and impact behavior are necessary before the designer can feel reasonably confident he has based material selection on sufficient information. A detailed discussion of the mechanical properties of materials and their incorporation into design can be found in Ref. 25.

Simple, unifying relations between these properties cannot be found since some vary widely over a number of parameters. For example, the ultimate strength of an element manufactured of a given material can vary greatly with its heat-treat condition, its size, its shape, and with the production method used in its fabrication. An interesting relation exists between modulus of elasticity (or rigidity) and density. This ratio, modulus of elasticity/density and known as the specific stiffness, is reasonably constant for the four metal families used in most military vehicle structures—steel, aluminum alloys, magnesium alloys, and titanium alloys. Fig. 4-4 shows these relationships as straight-line plots with maximum deviation bands of approximately ±9 and ±7 percent, respectively, for elasticity and rigidity moduli. These deviations include the effects of the factors heat-treatment, geometry, and production methods that were mentioned earlier. Better correlations based on material properties alone are possible. Of course, there are many exceptions to this simple comparison when a wide range of materials is examined for specific stiffness as in Fig. 4-5. The materials covered by Fig. 4-4 are indicated on Fig. 4-5.

The linearity of Fig. 4-4 indicates that modulus of elasticity (or rigidity) alone is not sufficient for comparative material selection. One alternative is to examine some basic strength-of-materials equations as functions of material properties to arrive at comparative results. Eqs. 4-22 through 4-25 can be used for this purpose.

BASIC EQUATIONS

Initial Form — *Secondary Forms*

$$S \propto \frac{1}{Z} \rightarrow \frac{S_1}{S_2} = \frac{Z_2}{Z_1}, \quad \text{beam or plate strength} \tag{4-22}$$

$$S \propto E\left(\frac{t}{b}\right)^2 \rightarrow \frac{S_1}{S_2} = \left(\frac{t_1}{t_2}\right)^2 \left(\frac{b_2}{b_1}\right)^2, \quad \text{column strength} \tag{4-23}$$

$$S \propto \frac{1}{EI} \rightarrow \frac{\delta_1}{\delta_2} = \frac{E_2 I_2}{E_1 I_1}, \quad \text{beam deflection} \tag{4-24}$$

BASIC EQUATIONS (Cont'd)

Initial Form	*Secondary Forms*		
$S \propto \frac{1}{\delta}$	$\longrightarrow \frac{\delta_1}{\delta_2} = \frac{D_2}{D_1} \approx \frac{E_2}{E_1}\left(\frac{t_2}{t_1}\right)^3$,	plate deflection	(4-25)

where

S = normal stress, lb/in.^{2}

E = modulus of elasticity of material, lb/in.^{2}

I = moment of inertia of a section under load (about an axis through its centroid and perpendicular to the load), in.^{4}

$Z = \frac{I}{y}$ section modulus, in.^{3}

y = perpendicular distance from centroidal to outer fiber of section, in.

b = width of column (dimension perpendicular to load), in.

t = thickness of members (dimension parallel to load), in.

δ = deflection, in.

$D = \frac{Et^3}{12(1-\mu)^2}$, flexural rigidity in plates (analogous to term EI in beams), lb-in.

μ = Poisson's ratio, dimensionless.

Subscripts 1 and 2 refer to separate cases being compared.

These equations show that it is impossible to separate material properties from geometric or form properties, and the latter give rise to weight considerations. Therefore, simultaneous material selection and shape selection are required to assure proper weight, strength, and stiffness in automotive structures. Eq. 4-22 to 4-25, and others of this type written for different loadings and different structural shapes, can be used in two ways for making preliminary material selections—(1) to predict changes in strength or deflection resulting from changes in material or form properties (by means of the initial forms of the equations), and (2) to make useful comparisons of the strength or deflections of different materials of the same or different weights by means of the second forms of the basic equations. Because Poisson's ratio appears in many basic structural relationships (apparent in Eq. 4-25, but not so in Eq. 4-23, where it affects the proportionality factor), use of these types of equations is recommended only when comparing like materials (metal with metal, plastic with plastic). At least one other restriction applies to these basic relations; the stress levels must be below the proportionality limit of the material. Some design stresses in metal structures exceed that limit and, in such cases, corrected formulas or plasticity corrections (see par. 4-2.3.3.2) based on specific stress-strain curves are necessary. For comparative purposes, the proportionality limit can be exceeded; and S_1/S_2 can be the ratio of either the yield point or ultimate stresses.

Either strength or deflection is usually the criterion for design; and on either basis, materials with high strength-to-weight ratios have the advantage when minimum weight is a requirement. Materials with higher moduli of elasticity show an initial advantage through higher yield and ultimate strengths and, thereby, higher allowable stresses. If high strength is obtained by resorting to the use of a relatively heavy material, the initial disadvantage of greater weight can be offset by employing more efficient structural shapes—i.e., improved section modulus for strength and greater moment of inertia or thickness for deflection—at a savings in weight. Eq. 4-24 can be used to arrive at the same results. Similar weight savings occur when strength is the design criterion. As an illustration, if a steel beam and an aluminum beam of the same span and of a geometrically similar cross-sectional form are to have the same weight, their section moduli will be proportional to their thicknesses and inversely proportional to their densities. This will result in a strength ratio of approximately 9:1. Eq. 4-22 shows this to be equivalent to a computed working (or other comparable) stress ratio. The actual strength advantage that most steels have over aluminum is not nearly this high, however. Thus, the aluminum beam can be lighter (smaller cross section) by a percent difference between the computed and actual strength advantage of steel to aluminum. These principles have been widely adopted in the aircraft industry and, more recently, are being adopted in the automotive industry.

Material selection based on the foregoing high strength-to-weight ratio principle is not always the only or final consideration for military vehicles. Better fatigue, impact, or wear

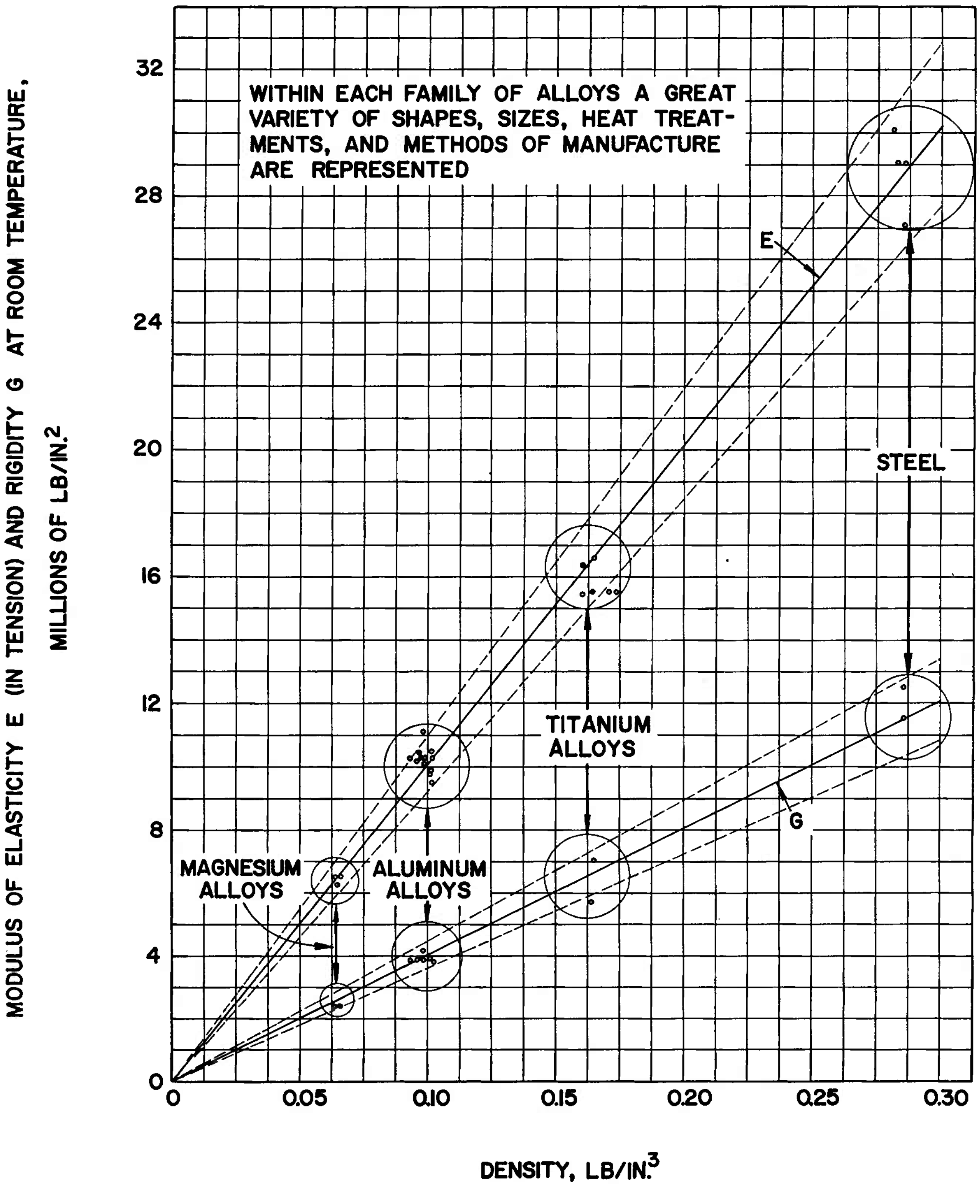


Figure 4-4. Relation of Density to Moduli of Elasticity and Rigidity for Selected Materials

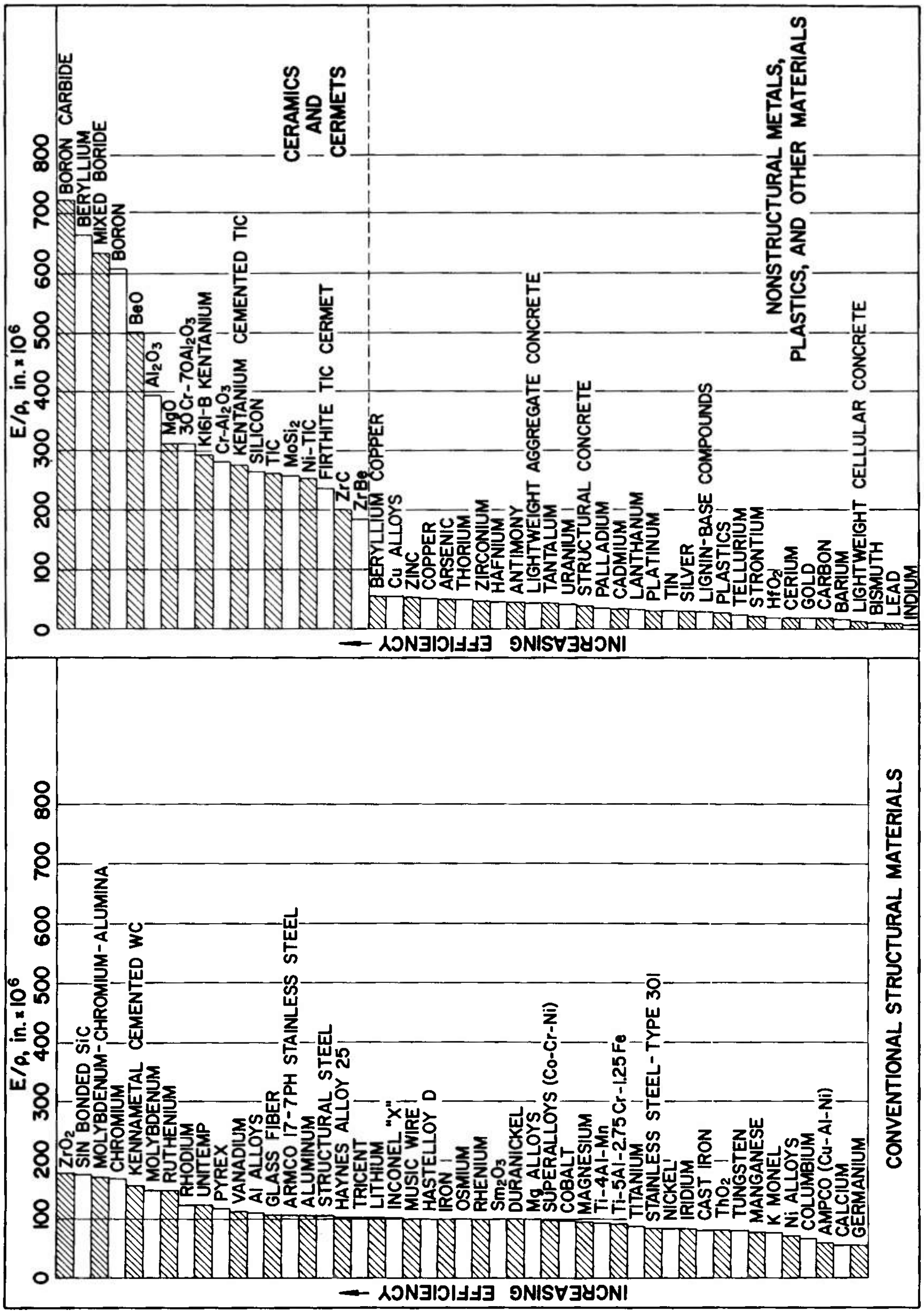


Figure 4-5. Specific Stiffnesses of Various Materials [26]

resistance; environmental factors; and cost can swing final selection to materials of greater than minimum weight.

The publications of material manufacturers are excellent aids to materials selection. Additional specific information sources on materials are Refs. 6, 27 through 29, and 168 through 171, while lists of numerous military specifications on automotive materials can be found in Ref. 30.

4-2.2 SHAPE SELECTION

Basic strength-of-materials relations are combinations of material and geometric parameters, which, as pointed out in the preceding paragraph, must be determined simultaneously. Eqs. 4-22 to 4-25 are typical of the basic relations, and functions of shape parameters in general are discussed in par. 4-2.1.

The most important shape selection concerns cross sections of structural members which, in terms of the more exact shape parameters—moment of inertia I and section modulus A—prescribe how well loads and deflections, both transverse and torsional, are resisted in those members. Physically, the moment of inertia of an area A with respect to an axis passing through the centroid of the area is the integral of the square of the perpendicular distance y of each elemental area dA from the selected axis times the elemental area dA, i.e.,

$$I = \int y^2\, dA$$

Obviously, there can be as many values of I as there are axes chosen; however, in most structural work, moments of inertia with respect to an orthogonal coordinate system—$I_{x\text{-}x}$, $I_{y\text{-}y}$ for transverse loading and $I_{o\text{-}o}$ for torsional loading, as shown in Fig. 4-6(A)—are all that is needed. Hence,

$$\left.\begin{aligned} I_{x\text{-}x} &= \int_A y^2\, dA \\ I_{y\text{-}y} &= \int_A x^2\, dA \\ I_{o\text{-}o} &= \int r^2\, dA = J, \text{ polar moment of inertia} \end{aligned}\right\} \quad (4\text{-}26)$$

$$\left.\begin{aligned} Z_{x\text{-}x} &= \frac{I_{x\text{-}x}}{y_{(max)}} \\ Z_{y\text{-}y} &= \frac{I_{y\text{-}y}}{x_{(max)}} \\ Z_{o\text{-}o} &= \frac{I_{o\text{-}o}}{r_{(max)}} \end{aligned}\right\} \quad (4\text{-}27)$$

where I and Z have dimensions of in.^{4} and in.^{3} , respectively, if x,y,r are expressed in in., and A in in.^{2} . When moment of inertia of a section about an axis other than that passing through its centroid is required (as is the case for cross sections resolvable into simpler shapes), Eqs. 4-26 may be used with distances x and y modified as given in Fig. 4-6 (B).

$$\left.\begin{aligned} I'_{x\text{-}x} &= \int_A (y + d_y)^2 dA = \int_A (y^2 + 2yd_y + d_y^2) dA \\ I'_{y\text{-}y} &= \int_A (x + d_x)^2 dA = \int_A (x^2 + 2xd_x + d_x^2) dA \end{aligned}\right\} \quad (4\text{-}28)$$

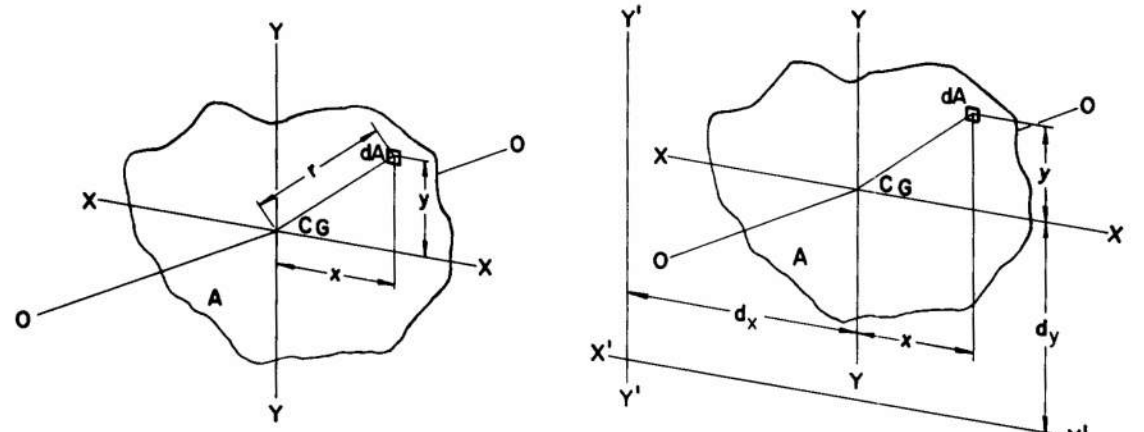
Reduction of Eq. 4-28 gives *parallel axis theroems* with which, if centroidal moments of inertia $I_{x\text{-}x}$ and $I_{y\text{-}y}$ are known, inertias $I'_{x\text{-}x}$ and $I'_{y\text{-}y}$ about any axes parallel to the centroidal may be calculated.

$$\left.\begin{aligned} I'_{x\text{-}x} &= I_{x\text{-}x} + Ad_y^2 \\ I'_{y\text{-}y} &= I_{y\text{-}y} + Ad_x^2 \end{aligned}\right\} \quad (4\text{-}29)$$

Solutions of Eqs. 4-26, 4-27, and 4-29 for numerous practical cross-sectional shapes appear in many engineering handbooks. Eqs. 4-26, although not useful in the basic forms shown, illustrate the underlying principle of efficient shape selection. These equations show that the contributions of elemental areas to moment of inertia increases with distance of such elements from the centroidal axis. Therefore, the increase in moment of inertia that results from transferring material from locations at or near the centroid to outer portions of a section is much more rapid than the accompanying weight increase. Depending on degree of symmetry, moment of inertia will be a function of either the third or fourth power of shape dimensions, while weight will be either a linear or square function of the same dimensions.

Such manufacturing processes as rolling, drawing, and extruding make the foregoing concept practical. The extrusion process, especially in aluminum manufacturing, has

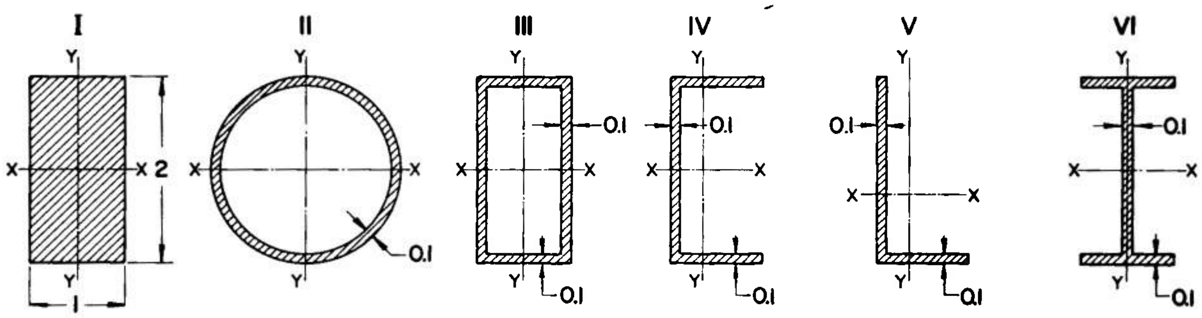
reached such development that increased analysis by the designer with regard to shape selection is worthwhile. The moments of inertia of some commonly used "efficient" structural shapes are compared with solid rectangular section in Fig. 4-6(C). No particular attempt was made to optimize the dimensions of the formed shapes; for simplicity, outer dimensions were kept the same as those of solid section. Effects of fillets and tapers were neglected. Greater advantages would be obtained if the outer dimensions were increased so that formed sections would more nearly equal the weight of solid sections. Even this quick comparison (Fig.



(A) WITH RESPECT TO CENTROIDAL AXES

(B) WITH RESPECT TO NONCENTROIDAL AXES

PHYSICAL INTERPRETATION OF MOMENT OF INERTIA



SECT.	I_X	I_Y	J	W*	I_X/W †	I_Y/W †	J/W †
I	0.667	0.167	0.834	2.00	1.00	1.00	1.00
II	0.270	0.270	0.540	0.596	1.36	5.40	2.17
III	0.280	0.090	0.370	0.560	1.50	1.92	1.58
IV	0.230	0.036	0.266	0.380	1.82	1.13	1.68
V	0.131	0.022	0.153	0.290	1.36	0.905	1.26
VI	0.230	0.0168	0.247	0.380	1.82	0.524	1.54

* WEIGHT/UNIT LENGTH
† VALUES IN LAST THREE COLUMNS ARE RELATIVE TO I/W OF SECTION I

(C) COMPARISON OF VARIOUS SHAPES

Figure 4-6. Moment of Inertia Considerations

4-6(C)) points out several shape selection guidelines, however, if types and directions of loads are reasonably well known:

a. Solid sections are generally inefficient because of their weight, as substantiated by the values in the last three columns of Fig. 4-6(C), but offer advantages in tension members because of their greater cross-sectional areas.

b. Asymmetrical sections (angles) or sections symmetric about one axis (I-beam, channel, rectangular box section) are well suited to resist transverse loads applied in one direction. They must, of course, be oriented with their depths in the direction of load (y-direction, Fig. 4-6(C), by their greater values for I_x and I_x/W than for I_y and I_y/W.

c. Members of the type cited in b. become progressively less ideal when the loading axis rotates from y to x; minimum usefulness occurs when the load is along axis x. For example, the I-beam is the least efficient of the shapes shown when loaded in the x-direction. (Compare I_x and I_y.) Closed sections (e.g., rectangular box, elliptic) offer reasonable strength in two or more directions, depending on the ratio of their principal outer dimensions.

d. Closed sections symmetric about two axes (e.g., circular, square box) are effective for loads applied from any direction. (Compare I_x and I_y for Shape II.)

e. Closed sections (I, II, and III) also have favorable torsional resistance properties (compare J values). Such properties generally decrease with the more open sections (IV, V, and VI). Ratios of polar moment of inertia to weight (J/W column of Fig. 4-6(C)) substantiate this. Except for the given dimensions, the channel shows a slight advantage over the box section.

Square and rectangular section tubing is commonly used in automotive structures not only because of excellent strength-to-weight ratios but also because of ease of fabrication, i.e., joining to flat rather than, say, round surfaces as shown in Fig. 4-7[31]. Shapes of interest to military vehicle structures designers will be those of Fig. 4-6(C), their combinations, or slight variations. Except for some sandwich configurations (par. 4-2.3.2), special shapes produced by extrusions, or lightweight castings (see Sections II, III, and IV of this chapter), elaborately built-up or complex shapes are neither economical nor competitively efficient. Some insight into shape effectiveness is available from a series of tests comparing an unstiffened flat plate with variously stiffened plates[32]. Results of the tests, Table 4-4 and Fig. 4-8 support the use of common shapes.

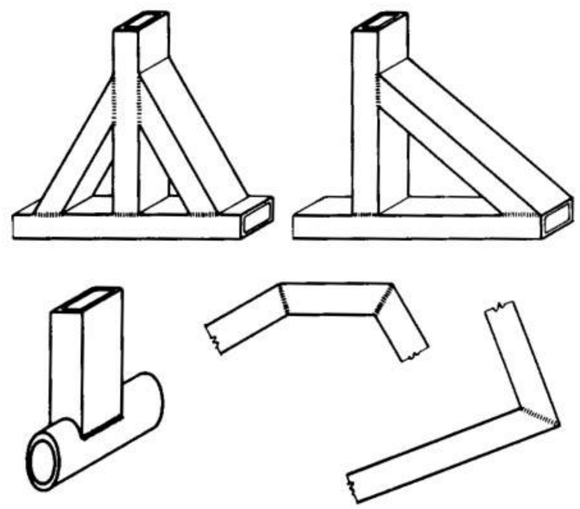


Figure 4-7. Shape Selection for Ease of Fabrication[31]

Again, publications of material manufacturers offer valuable guidance for shape selection, and describe new or increased production capabilities. Refs. 33 and 34 contain useful information and illustrations of shape selection for minimum weight in vehicle structures. Also see Ref. 26 for interesting new developments about combined material and shape selection.

4-2.3 LIGHTWEIGHT CONSTRUCTION TECHNIQUES

Lightweight construction techniques strive to increase the strength-to-weight ratio of a structure primarily by optimum weight reduction. This can be accomplished by lightening holes, scarfs, novel fabrication of load-bearing members (notably the various sandwich panels), by employing a special class of construction (e.g., stressed-skin structures that remain effective after local buckling (par. 4-2.3.3)), or by combinations of these techniques.

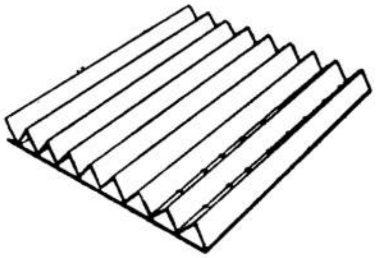
4-2.3.1 Use of Lightening Holes, Chamfers, and Scarfs

Many structural building blocks weigh more than necessary to supply the required strength. Excess weight may be due to general overdesign

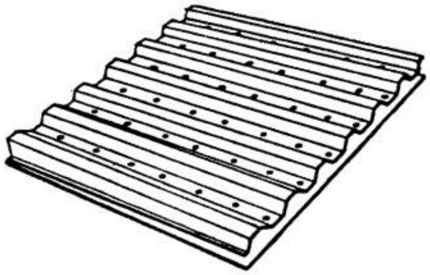
TABLE 4-4 SHAPE EFFECTIVENESS IN STIFFENERS[32]

Load	Application	Stiffener Shape Number Refer to Fig. 4-8														
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Torsion	Single	48	40	28	11.3	3.5	27	33	24	1.3	10.6	1.3	4.2	3.3	7.5	1.6
	Sandwich	62	29	20	8.1	29	85	44	42	48	24	4.8	18	56	21	8.7
Bending (single)	x-Axis	112	58	18	47	10	10.3	12.5	22.5	16.3	8.8	1.0	1.6	27	7.5	1.2
	y-Axis	1.1	1.3	30	1.2	10	10.3	12.5	7.8	16.3	8.8	1.0	9.3	6.9	65	1.2
Bending (sandwich)	x-Axis	150	62	19	62	11	83	16	29	32	3.6	3.6	27	52	30	6
	y-Axis	56	13	16	2.0	11	83	16	29	32	3.6	3.6	7.3	31	10	6
$\frac{\text{Strength}}{\text{Weight}}$	Single, torsion	30	26	17	8.1	2.1	10	18	1.5	7.3	6.8	1.0	2.6	2.1	4.1	1.1
	Single, bending	x 72	28.2	10	34.5	5.8	3.4	6.7	13	9.1	5.7	0.74	1.1	16	3.9	1.1
		y 0.75	0.8	18	0.85	5.8	3.4	6.7	4.6	0.1	5.7	0.74	5.4	4.2	34	1.1
	Sandwich, torsion	48	23	14.5	6.5	23	45	32	32	33	19	40	14	44	15	7.3
	Sandwich, bending	x 116	50	14	50	8.5	44	11	22	21	2.6	2.6	21	41	21	5
		y 43	10	12	2.3	8.5	44	11	22	21	2.6	2.6	5.6	24	7	5

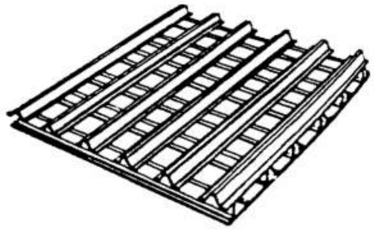
LEGEND Stiffness value of the reference plate is taken to be 1, and values for the other plates indicate how many times stiffer they are than the reference. All plates were 500-mm square (20-in.) and 3-mm (about 1/8-in.) thick. Web thickness was 1-mm (about 1/32 in.); height was 40-mm (about 1-5/8 in.). All constructions were welded.
For sandwich construction the reference-plate thickness is doubled to give a thickness of 6-mm.



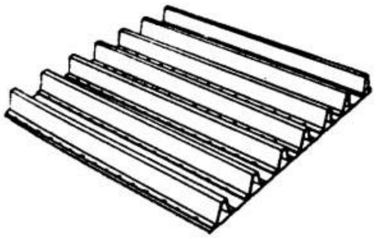
1 Angles



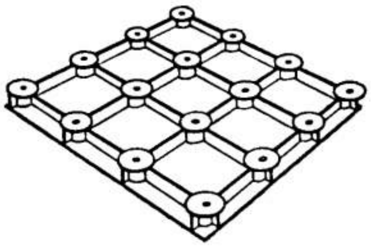
2 Corrugated plate



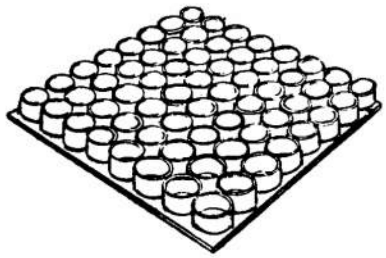
3 Two layers of single channels arranged crosswise



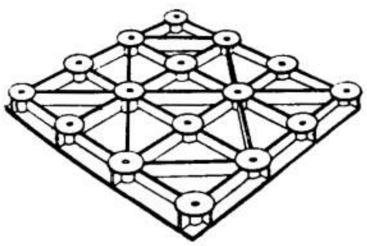
4 Single layer of channels



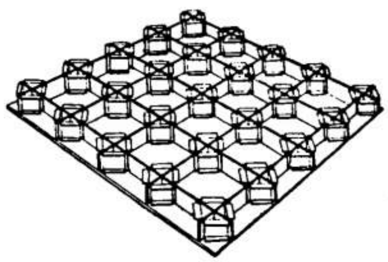
5 Round gussets and straight ribs



6 Tube sections welded to each other

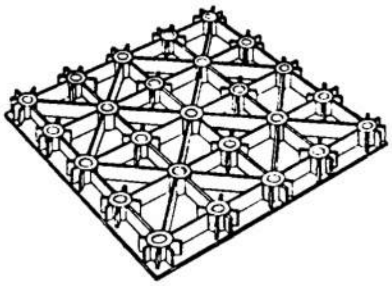


7 Round gussets, straight and diagonal ribs

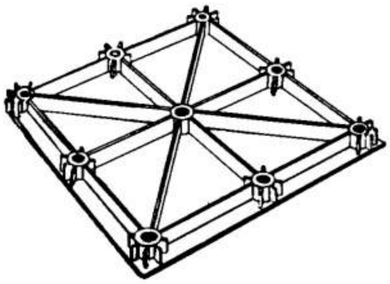


8 Gussets made from channel sections

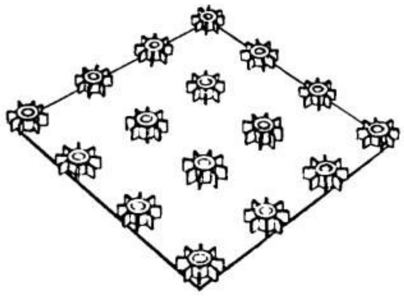
Figure 4-8. Shape Effectiveness in Stiffeners[32] (1 of 2)



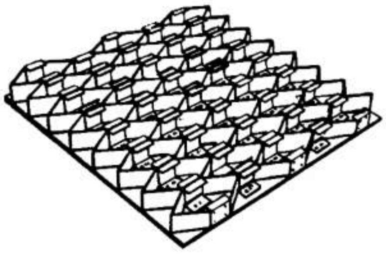
9 Starlike gussets and radial ribs (close together)



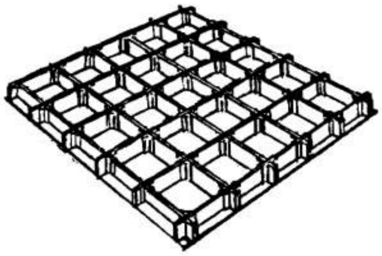
10 Starlike gussets and radial ribs (far apart)



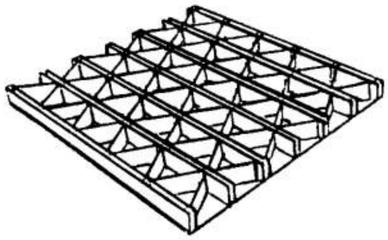
11 Starlike gussets without ribs



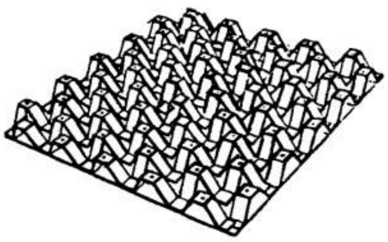
12 Honeycomb



13 Straight sheet ribs and angles



14 Studs, strips and angles



15 "Weave" pattern

Figure 4-8. Shape Effectiveness in Stiffeners[32] (2 of 2)

or lack of analysis for "lightly loaded" areas in a given member. Simple manufacturing methods dictated by economic considerations also contribute to the overweight problem. Possibly such designs are allowable in some vehicle structures, but certainly not in modern lightweight military vehicles. Use of lightening holes, chamfers, and scarfs is helpful in reducing excess structure weight.

Lightening holes are openings in a structural member added during or after manufacture. On completed members, the openings may be burned, sawed, bored, etc. Cost considerations enter the picture again because any method selected represents additional work on each piece before it is fully usable—the simplest production method is, of course, preferred. Lightening holes in stamped or cast members are part of initial design and, thus more efficient; here increased initial costs are absorbed over the many pieces produced. Automotive lightening holes appear in webs of beams, tank bulkheads, and various mounting bracket parts. Specific examples can be found in sides of box section crane booms, support brackets for side plates of tracked vehicle fenders (Figs. 1-15, 1-16, and 1-26), and in a chassis-to-body (platform) mounting bracket for a lightweight vehicle shown in Fig. 4-9. Lightening holes in thin-walled stampings are often supplemented with curled or flanged hole edges to make up for loss of section.

Formation of chamfers is a means of eliminating sharp edges formed by the intersection of a curved surface with a flat surface on a part, e.g., removable guide pins, dump body pivots, and various studs. Since chamfering is accomplished by material removal, weight is also reduced; although the amount of weight removed in this is exceedingly small. In addition, the application of chamfers in the structures under consideration is limited.

Scarfs, as applicable here, refer to weight reduction by moving material from "nonworking" areas of parts. Strictly put, this method describes some machining operation after manufacture of a given part or its components since similar cutouts in, for example, stampings or die-castings are produced at the time the basic part is produced. Therefore, the cost considerations relative to lightening holes are also applicable to scarfs. Many examples of the scarf can be found in an automotive body or hull; an example concerning a typical mounting bracket is offered in Fig. 4-10, where dead corners of the support plates and gusset are clipped off.

In most cases material removal, especially by scarfs and lightening holes, cannot be made without performing some stress analysis on the member.

4-2.3.2 Designing With Sandwich Panels[39-41]

A sandwich panel is a lamination of various materials bonded together into a single unit having a high strength-to-weight ratio. In general, it is formed with an inner *core* flanked by a layer of adhesive and an outer layer of *facing skin* on either side. The inner core is made from a low-density material such as styrofoam or paper, the facing skin from a high-density material such as carbon steel or aluminum alloy. This allows the best of two worlds—high strength without great weight.

The varieties of sandwich possible are limited only by a designer's imagination and fabrication feasibility. Various low-density solid cores exist, but the efficiency of this construction technique (high-density skin, low-density core) is better realized in the many types of very low-density, nonsolid cores. Fig. 4-11 shows some representative sandwich panels.

Honeycomb sandwich is particularly notable in the high-strength-versus-weight property and production methods to fabricate its hexagonal cellular core are well established. The honeycomb sandwich is a versatile structural material; it can act as beam, column, or plate, depending on its dimensions and orientation to loads. Substantial overall thickness of the panel is an important parameter for the designer since core thickness can be increased (Table 4-5) with very little weight increase; this contributes to higher load capacity and stability. When the material is loaded as a beam, the facing skins at extremes of the neutral axis take all tensile and compressive (bending) loads and transmit shearing forces through the adhesive bonding to the core centered about the neutral axis. As Fig. 4-12 shows, the ratio of strength to weight in bending is favorable compared to that of other constructions. When loaded as a column, sandwich skins can take all the end loading directly because of continuous core bracing against buckling[35].

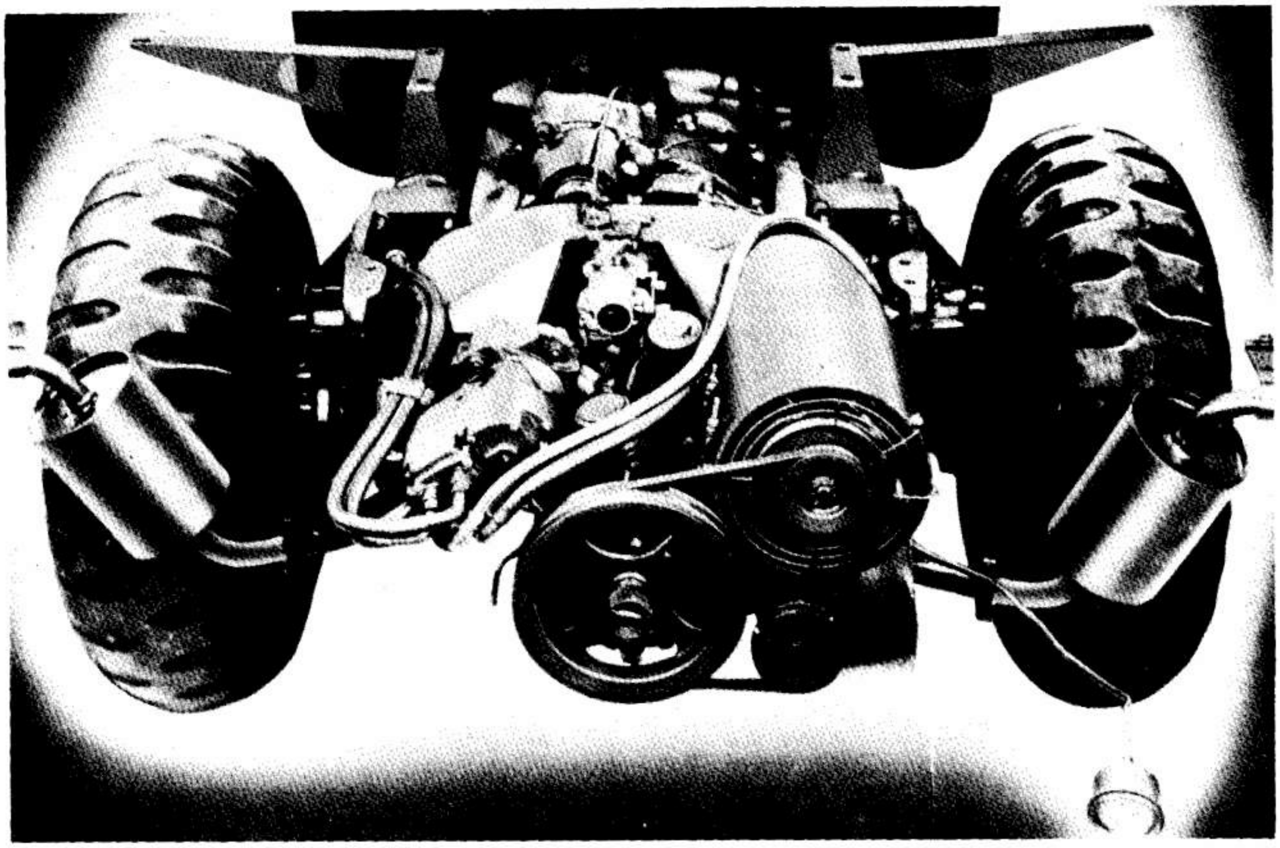


Figure 4-9. Application of Lightening Holes in Light Weapons Carrier, M274

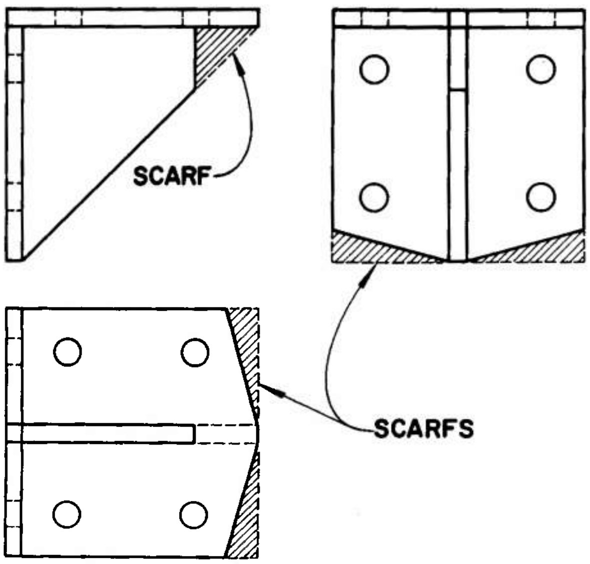


Figure 4-10. Use of Scarfs and Mounting Bracket

The honeycomb core geometry is produced by expanding a stack of alternate layers of continuous core strips and adhesive segments after curing. Cell shape is varied by controlling the expansion process (Fig. 4-13(A), (B), and (C)). Adhesive spacing variations within the stack produce additional shapes upon expansion, as Fig. 4-13(D) and (E) shows. Materials, gages, and adhesives not adaptable to expansion are produced from corrugated core strips joined mainly by welding, although bonding is sometimes used[36]. This process makes still other cell shapes possible (Fig. 4-13(F)).

Honeycomb is orthotropic, i.e., its properties differ with direction in material. For example, both shear strength and modulus of rigidity (shear modulus) are substantially higher (on the order of 2:1)[37] in a direction along the core strip than in a direction perpendicular to the strip (directions L and W, respectively, in Fig. 4-13; these are in accordance with MIL-C-8073A and MIL-C-7438C). These two properties also

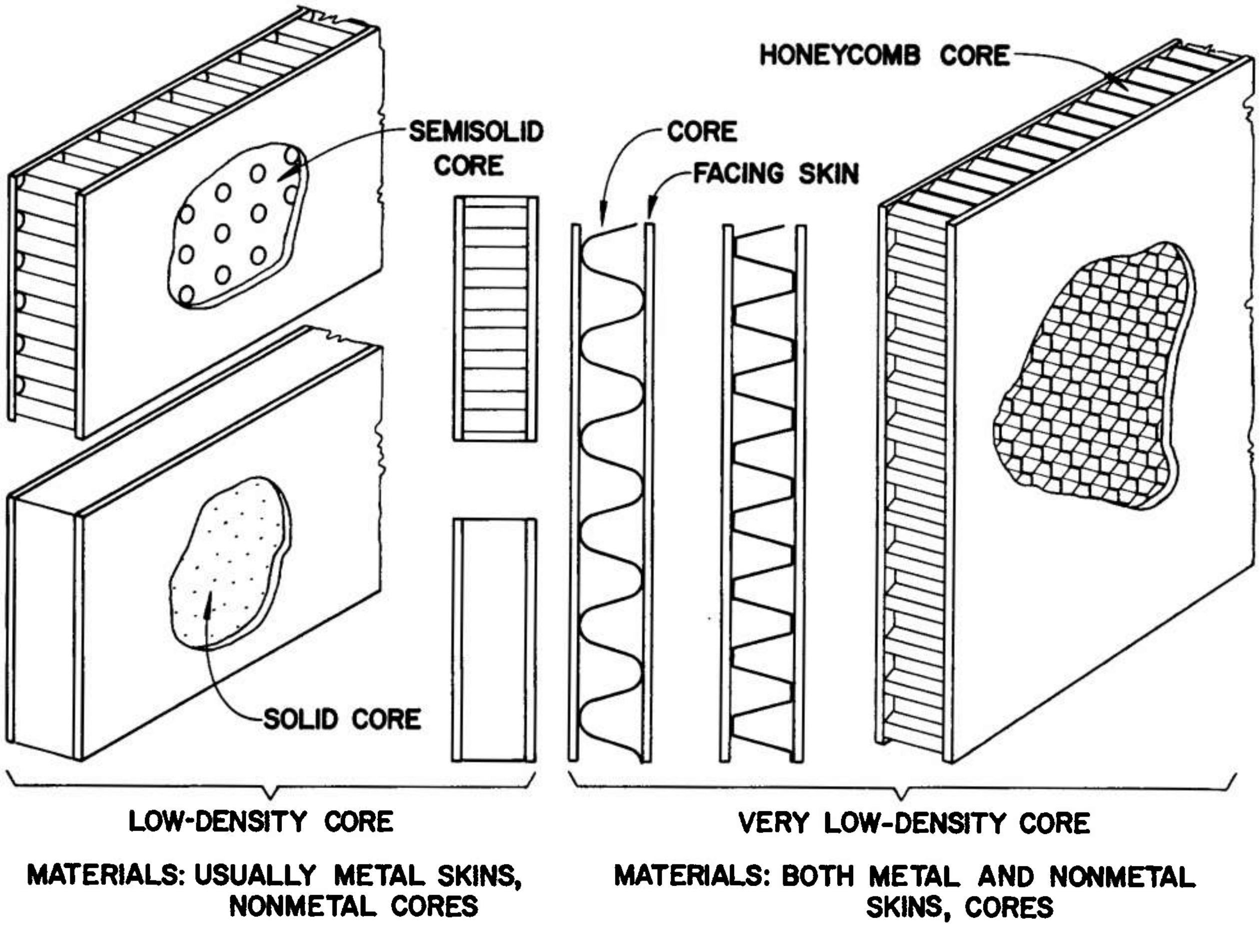
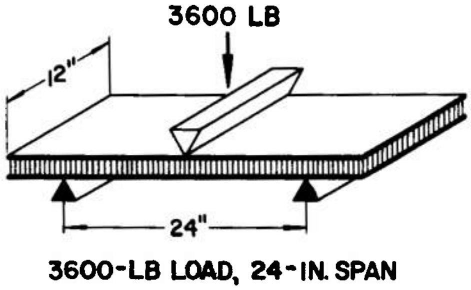


Figure 4-11. Representative Sandwich Panels

vary with core density; Fig. 4-14 shows typical shear strength and modulus variation for aluminum honeycomb, in a direction along the core strip. Typical core variables and size data are given in Table 4-5.

The material variation in facing skins is great, but thickness is limited to small values in sandwich construction. Examples of facings that have been used on heavy duty flooring and highly stressed panels (thereby applicable to military conditions) are carbon steel 0.030 to 0.050-in. thick and 7075-T6 aluminum alloy 0.010 to 0.093 in.-thick[36]. Mechanical properties of typical facing skins are given in Table 4-5.

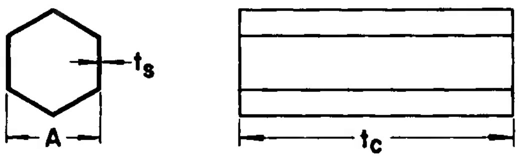
The honeycomb sandwich must be protected from concentrated loads. These cannot always be foreseen, but for suspected areas of high loading (e.g., cargo tiedowns, rivet or bolt connections between panels) inserts between facing skins are used to distribute loads. Inserts may be prefabricated from various materials and can take many forms; prefabricated inserts are usually bonded to the core before final panel assembly, but may be added afterward in some cases. Fasteners may also be bonded directly to the sandwich by filling cells with an epoxy or similar curable material into which the fasteners are placed; these plastic inserts are added after completion of a panel by careful drilling through one facing skin (or through recesses planned for this purpose). Fig. 4-15 shows several representative sandwich inserts. Parts (A) and (B) are plastic inserts for bonding fasteners or attachments. The remaining parts of the figure are prefabricated insert types—(C) and (D) use extruded shapes, (E) and (F) accept bolts and also use locally increased facing thicknesses for load distribution, (G) is a threaded insert pressed into the completed panel, (H) through (K) are varieties of solid inserts with and without attachments.



MATERIAL	DEFLECTION, IN.	WEIGHT OF PANEL, LB
HONEYCOMB SANDWICH	0.058	7.79
NESTED I-BEAMS	0.058	10.86
STEEL ANGLES	0.058	25.9
ALUMINUM PLATE	0.058	34.2
MAGNESIUM PLATE	0.058	26.0
STEEL PLATE	0.058	68.6
GLASS-REINFORCED PLASTIC LAMINATE	0.058	83.4

Figure 4-12. Comparative Bending Strength of the Honeycomb Sandwich[36]

TABLE 4-5. TYPICAL CORE DATA



VARIABLE	TYPICAL DIMENSIONAL RANGE
CELL SIZE A	1/32-TO 3-in. (1/8-TO 1-1/2-in. STANDARD PRODUCTION)
CORE THICKNESS t_c	0.060 TO 24 in.
STRIP THICKNESS t_s	0.0007 TO 0.005 in.
CORE DENSITY	2 TO 12 lb/ft^3
OVERALL SIZE, ONE PIECE	TO 48x96 in.

Edges of the honeycomb panel, as normally manufactured, are not protected. To provide for edge loading, build up panels larger than provided by one-piece core sizes (Table 4-5), and generally protect the core. Edge inserts in the form of curable putty, blocks, or extruded structural shapes are added by bonding, as shown in Fig. 4-16(A) through (H). The facing skins may also be formed in several ways to

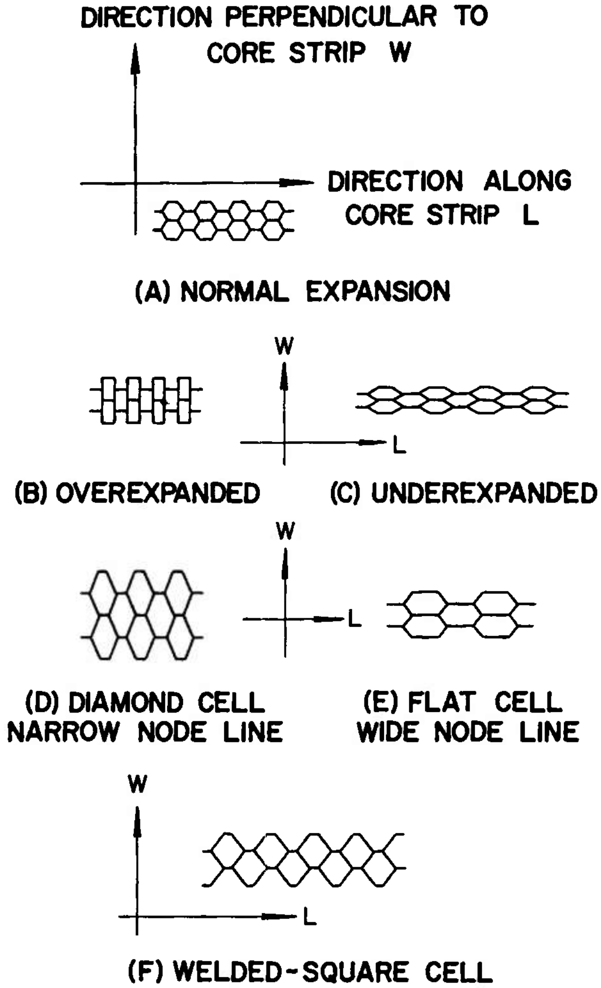


Figure 4-13. Cell Shape Variations of the Honeycomb Core[35]

protect panel edges (Fig. 4-16(I) through (L)). Combinations of techniques, including use of rivets, are shown in Fig. 4-16(M) through (S). Many edge insert types are particularly suitable for joining panels in one plane or two planes as in corners of enclosed sandwich structures. For the latter purpose, specifically, Fig. 4-17 offers additional detail. Ref. 42 gives additional information on inserts, edges, and joining details.

Joints in the honeycomb core itself are possible using adhesives similar to those used in the basic sandwich bonding. Such methods are well-developed and offer yet other ways of producing panels larger than the size limit set by standard cores. Another purpose for core joints is to obtain a panel of variable density—denser core areas being located under high loads. Fig.

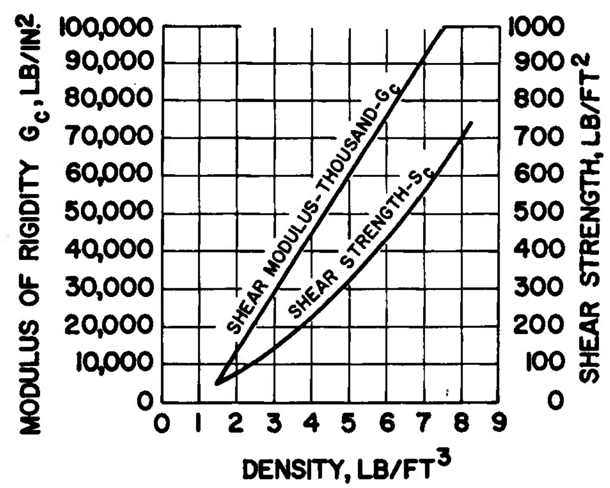


Figure 4-14. Typical Aluminum Honeycomb Core Strength (L-direction) vs Density[36]

4-18(A) shows a butt adhesive joint used for the latter purpose. This much used and reliable joint type requires no special orientation of cells between members. The crush splice is also commonly used and offers economy; here, two pieces of honeycomb are overlapped (approximately a distance of two cells) and then simply crushed together[36]. Full shear strength of the core is developed by some adhesive placed in the crushed region. The beveled crush-splice (Fig. 4-18(B)) is used for joining thicker cores; the beveling (30° to 60°), increasing with thickness, is used to limit the crushing action to retain shear strength of core. Other core joint types require more exact orientation between cells of the two members. These are used in aircraft structures or in applications where continuity of the core is essential.

The honeycomb sandwich material offers several additional properties of interest to the military vehicle designer. The sealed, cellular structure has inherent buoyancy, and it tends to absorb vibration or other energy. In addition, nonmetallic sandwiches offer noise and environmental insulation qualities, and corrosion resistance. Ref. 43 offers additional information on the state of the art of honeycomb materials.

Many design procedures are available for sandwich panels but, at present, these are based on idealized edge support conditions. Treatment of combined loadings[6,44] is not as readily available as it is for more conventional construction techniques. Knowledge of edge support conditions is essential because connections to other members can take place along two edges of substantial dimension. In existing theories, only the cases of fixed, simple, or free (no) support are examined (also see par. 4-2.3.3.2 for edge supports). Even with such simplifying assumptions, the resulting "exact" solutions[45,48] of sandwich strength problems are complex and cumbersome. For these reasons, the use of a simplified analysis offers advantages.

Refs. 49 and 50 present one simplified method in which honeycomb sandwich panels are treated according to their orientation to load:

a. Load normal to facing skin:
   (1) various simple and fixed supports on one or two edges
   (2) fixed supports on all edges

b. End loading of facing skins, loaded edges simply supported, other edges free

c. Loading in plane of facing skins, parallel to edges (shear) all edges fized

Conditions a and c are encountered in panels acting as beams or plates, while b occurs in sandwiches used as columns.

For panels satisfying condition a, three important design formulas are of the form

(1) *Maximum normal facing skin stress*

$$S_f = \frac{M_{(max)}}{t_c t_f} = C_1\left(\frac{qL^2}{t_c t_f}\right) = C_2\left(\frac{PL}{t_c t_f}\right), \text{ lb/in.}^2 \quad (4\text{-}30)$$

(2) *Maximum core shear stress*

$$S_{s(c)} = C_3\left(\frac{qL}{t + t_c}\right) = C_4\left(\frac{P}{t + t_c}\right), \text{ lb/in.}^2 \quad (4\text{-}31)$$

(3) *Maximum deflection*

$$\left.\begin{aligned} \delta_{(max)} &= C_5\left(\frac{qL^4}{D}\right) + C_5 C_6\left(\frac{qL^2}{t_c G_c}\right), \text{ in.} \\ &= C_T\left(\frac{PL^3}{D}\right) + C_2\left(\frac{PL}{t_c G_c}\right) \end{aligned}\right\} \quad (4\text{-}32)$$

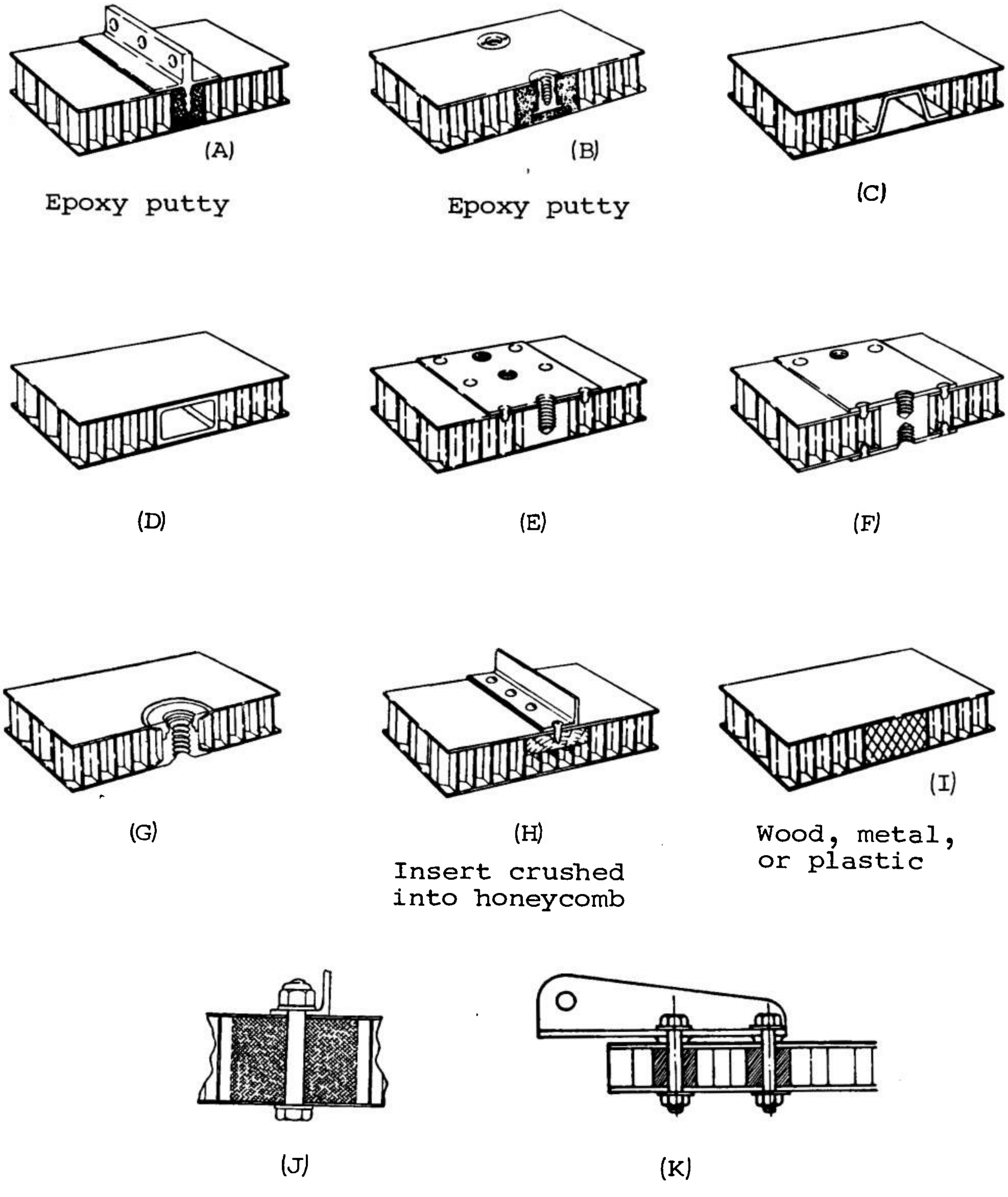


Figure 4-15. Typical Insert Details for Load Distribution and Attachments [36,50]

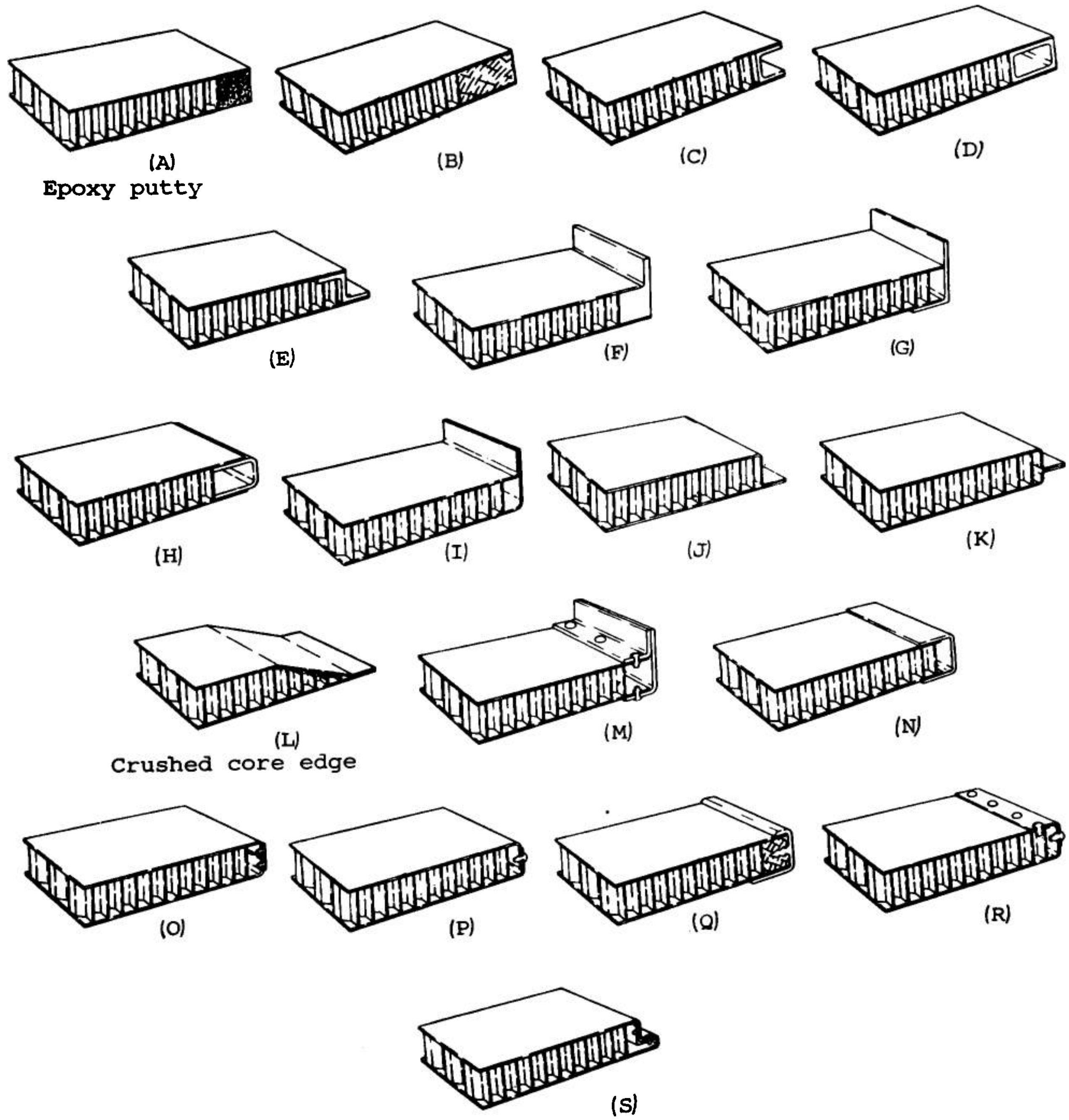


Figure 4-16. Typical Edge Details[50]

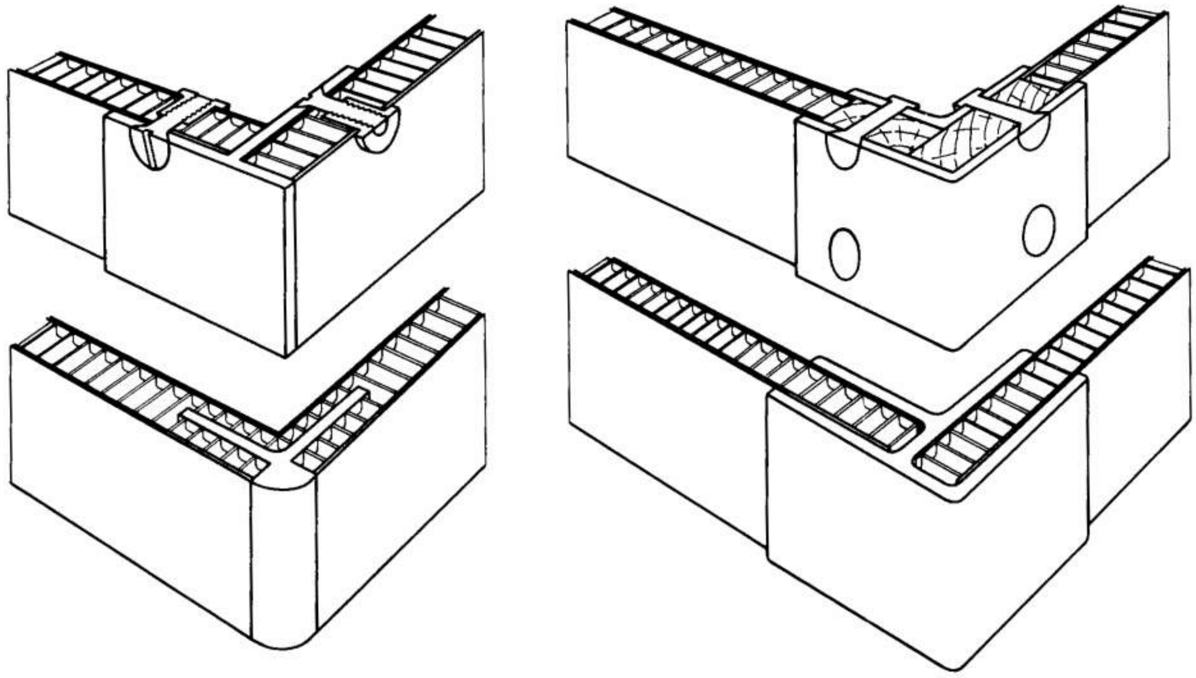


Figure 4-17. Typical Corner Details

where

$$D = \frac{1}{12(1-\mu^2)}\left[E_c t_c^2 + E_f (t^3 - t_c^3)\right], \quad (4\text{-}33)$$

exact formula

$$= \frac{E_f t_f t_c t}{2(1-\mu^2)}, \quad \text{approximate relation} \quad (4\text{-}34)$$

for both facing skins of same material and thickness

$$= \frac{E_{f1} E_{f2} t_{f1} t_{f2} (t + t_c)^2}{4(1-\mu^2)(E_{f1} t_{f1} + E_{f2} t_{f2})}, \quad (4\text{-}35)$$

approximate relation for facing skins of different material or thickness

$M_{(max)}$ = maximum bending moment for given loading and supports (Figs. 4-19 and 4-20), in.-lb/in. of width

q = distributed loading, lb/in./in. of width or psi

P = concentrated or total loading, lb/in. of width

t_c = core thickness, in.

t_f = facing skin thickness, in.

E_c, E_f = moduli of elasticity of core and facing skin, respectively, lb/in.^{2}

G_c = modulus of rigidity of core, lb/in.^{2}

C_1, C_3, C_5, C_6 = constants for distributed loading (Figs. 4-19 and 4-20), dimensionless

C_2, C_4, C_7 = constants for concentrated loading (Fig. 4-19), dimensionless

L = span length, in.

μ = Poisson's ratio for facing skin material (see Table 4-6), dimensionless

Solution of Eq. 4-32 is complicated by the presence of term D, which even in its approximate forms requires choice of core, facing, and sandwich thicknesses—unknowns of a given problem. In general, however, the second term of the deflection equation is very small relative to the first and, if neglected, an auxiliary relation for D is available to circumvent this problem when maximum deflection is specified

$$D = \frac{C_5 q L^4}{\delta_{(max)}} = \frac{C_7 P L^3}{\delta_{(max)}}, \text{ lb-in.} \quad (4\text{-}36)$$

TABLE 4-6 MECHANICAL PROPERTIES OF TYPICAL SANDWICH FACING MATERIAL[37]

Facing Material	*Yield Strength* S_{yield}, lb/in.^{2}	E_f, lb/in.^{2}	*Poisson's Ratio*		*Weight per mil thickness*, lb/ft^2	*Comments*
			μ	$1-\mu^2$		
Aluminum 1100-H14	17,000	10×10^6	0.33	0.89	0.0143	Low cost, weldable
Aluminum 2024-T4	47,000 42,000 (clad)	10.6×10^6	0.33	0.89	0.144	Combines good strength with reasonable cost
Aluminum 6061-T4	19,000	10.0×10^6	0.33	0.89	0.0141	Lowest cost, heat treatable, weldable
Aluminum 7075-T6	73,000 67,000 (clad)	10.4×10^6	0.33	0.89	0.0145	High strength
Titanium Ti-75A(annealed)	80,000	15×10^6	0.245	0.94	0.0235	
Titanium 6Al-4V (heat-treated)	143,000	16.8×10^6	0.245	0.94	0.0230	
Mild Steel 1015	45,000	30×10^6	0.30	0.91	0.0408	Low cost
Stainless Steel 316L	60,000	30×10^6	0.245	0.94	0.0410	Brazable
Stainless Steel 17-7PH	200,000	29×10^6	0.245	0.94	0.0397	High-strength
Reinforced-plastic 181 glass fabric at 90° epoxy resin	20,000	3.1×10^6	0.141	0.98	0.0094	
Reinforced-plastic 143 glass fabric at 0° epoxy resin	58,000	5.1×10^6	0.141	0.98	0.0094	
Reinforced-plastic 180 glass fabric at 90° polyester or phenolic resin	19,000	2.8×10^6	0.141	0.98	0.0094	Heat-resistant
Hardboard	300	0.4×10^6	0.10	0.99	0.0042	
Plywood Douglas fir	2,800	1.8×10^6	0.10	0.99	0.0025	

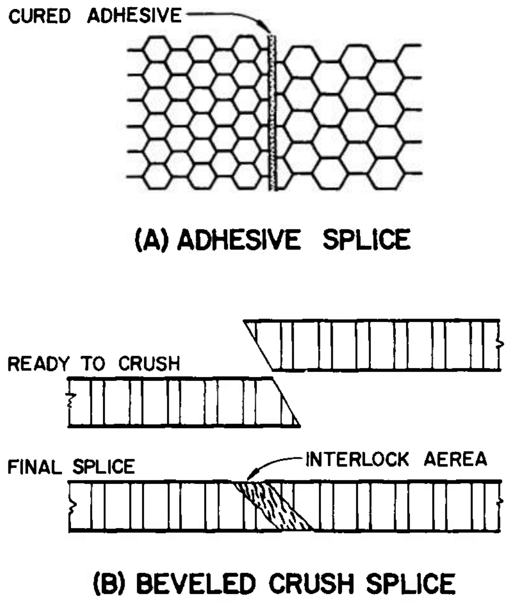


Figure 4-18. Core Joints[36]

Note that use of Fig. 4-19 is limited to sandwiches meeting the first instance of condition a and that *P and q must be expressed as loads per unit width of a quasibeam.* For the second instance of condition a, both panel length L and width b influence the governing equations; consequently the relation constants C_1, C_3, C_5, and C_6 are all functions of the ratio b/L, as shown in Fig. 4-20. Furthermore, different bending moments exist parallel to dimensions L and b, making it necessary to use $C_1{*}$ and C_1, respectively, to differentiate between these cases. Since $C_1 \geqslant C_1{*}$ for values of $b/1 \geqslant 1$ (Fig. 4-20), C_1 will be used for maximum calculations. Since the curves of Fig. 4-20 are based on the assumption that Poisson's ratio equals 0.3, they are applicable only for metals.

For condition a, further aid is available in the form of design curves that speed solution of Eqs. 4-30 to 4-32, give sizes of sandwich components, and provide for minimum weight analysis. An outline of steps for using the curves of Figs. 4-21 to 4-26 follows:

(1) Determine D from Eq. 4-36 and, using a flexural rigidity curve given in Figs. 4-21 to 4-24 for four different facing materials, find combinations of t_f and t.

(2) For t_c (from a chosen t_f and t), find $S_{s(c)}$ from Fig. 4-25 to satisfy maximum found in Figs. 4-19 or 4-20.

(3) Select core material with $S_{s(c)(min)} > S_{s(c)}$ actual; magnitude of $S_{s(c)}$ may be reduced if necessary by choosing a greater t_c. $S_{s(c)(min)}$ curves for some materials are given in Figs. 4-27 and 4-28, in terms of core density.) Final choice of cell size and strip thickness is based on core density.

(4) For a selected facing material (Table 4-6), allowable stress is taken as yield strength; actual stress S_f (limited by yield strength) is found in Fig. 4-26 for a given value of M (from Figs. 4-19 or 4-20) and C, where

$$C = t_c t_f, \text{ in.}^2 \tag{4-37}$$

with t_c and t_f as chosen in step 2.

(5) Weight optimization is now possible by using the thinner (core density parameter) curves on the flexural rigidity charts, at the original value of D (step 1).

(6) In general, increased values of t and t_c will result, leading to decrease of $S_{s(c)}$, S_f, and possibly of core density.

Thus, reiteration of steps (1) to (6) is needed, but final solution may be quickly accomplished by cyclic use of the curves. Checks must be made to ensure that limiting normal and shear stresses are not exceeded throughout the procedure.

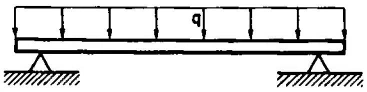
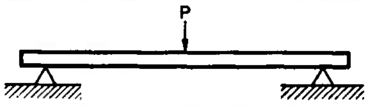
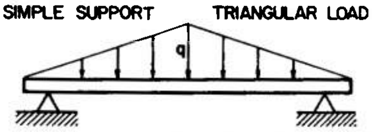
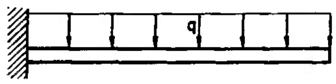
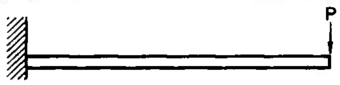
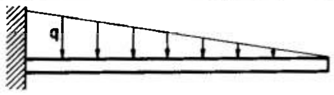
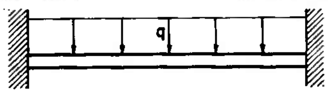
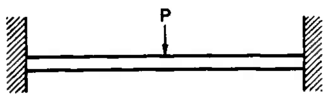
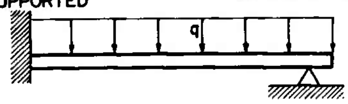
For condition b a column-loaded panel of height L and width b, and approximate relation[50] for compressive buckling stress of the facing skin is

$$S_{c(f)} = \frac{\pi^2 D}{2t_f \left(L^2 + \dfrac{\pi^2 D}{t_c G_c}\right)}, \text{ lb/in.}^2 \tag{4-38}$$

For condition c, a shear-loaded panel of dimensions L and b, the shear buckling stress of the facing skin is given by

$$S_{s(f)} = \frac{K\pi^2 E_f t_c t}{4(1-\mu^2)\, b^2}, \text{ lb/in.}^2 \tag{4-39}$$

where $K = 5.35 + 4(b/L)^2$, buckling coefficient, dimensionless. Other terms in Eqs. 4-38 and 4-39 are as defined for Eqs. 4-30 to 4-32. More detailed information on sandwich theory is found in Refs. 23 and 51. At this point, consider an illustrative example.

BEAM TYPE	EQUIVALENT TOTAL LOAD P, LB/IN.	MAXIMUM SHEAR FORCE $V_{(max)}$, LB/IN.	MAXIMUM BENDING MOMENT $M_{(max)}$, IN.-LB/IN.	FACING STRESS CONSTANTS C_1 AND C_2	CORE STRESS CONSTANTS C_3 AND C_4	BENDING DEFLECTION CONSTANTS C_5 AND C_6
SIMPLE SUPPORT UNIFORM LOAD 	qL	$\frac{qL}{2}$	$\frac{qL^2}{8}$	$\frac{1}{8}$	1.0	$\frac{5}{384}$
SIMPLE SUPPORT CENTER LOAD 	---	$\frac{P}{2}$	$\frac{PL}{4}$	$\frac{1}{4}$	1.0	$\frac{1}{48}$
SIMPLE SUPPORT TRIANGULAR LOAD 	$\frac{qL}{2}$	$\frac{qL}{4}$	$\frac{qL^2}{12}$	$\frac{1}{12}$	0.5	$\frac{1}{60}$
CANTILEVER UNIFORM LOAD 	qL	qL	$\frac{qL^2}{2}$	$\frac{1}{2}$	2.0	$\frac{1}{8}$
CANTILEVER END LOAD 	---	P	PL	1.0	2.0	$\frac{1}{3}$
CANTILEVER TRIANGULAR LOAD 	$\frac{qL}{2}$	$\frac{qL}{2}$	$\frac{qL^2}{6}$	$\frac{1}{6}$	1.0	$\frac{1}{15}$
ENDS FIXED UNIFORM LOAD 	qL	$\frac{qL}{2}$	$\frac{qL^2}{12}$	$\frac{1}{12}$	1.0	$\frac{1}{384}$
ENDS FIXED CENTER LOAD 	---	$\frac{P}{2}$	$\frac{PL}{8}$	$\frac{1}{8}$	1.0	$\frac{1}{192}$
ENDS FIXED AND SUPPORTED UNIFORM LOAD 	qL	$\frac{5qL}{8}$	$\frac{qL^2}{8}$	$\frac{1}{8}$	1.25	$\frac{1}{185}$

NOTE: $C_5C_6 = C_1$ WHEN USING MAXIMUM DEFLECTION EQUATION FOR DISTRIBUTED LOADS

$b=1 \begin{cases} P = \text{CONCENTRATED OR TOTAL LOAD ON A BEAM OF UNIT WIDTH} \\ q = \text{DISTRIBUTED LOAD ON A BEAM OF UNIT WIDTH} \end{cases}$

Figure 4-19. Beam Chart for Sandwich Panels Supported on One or Two Edges[49,50]

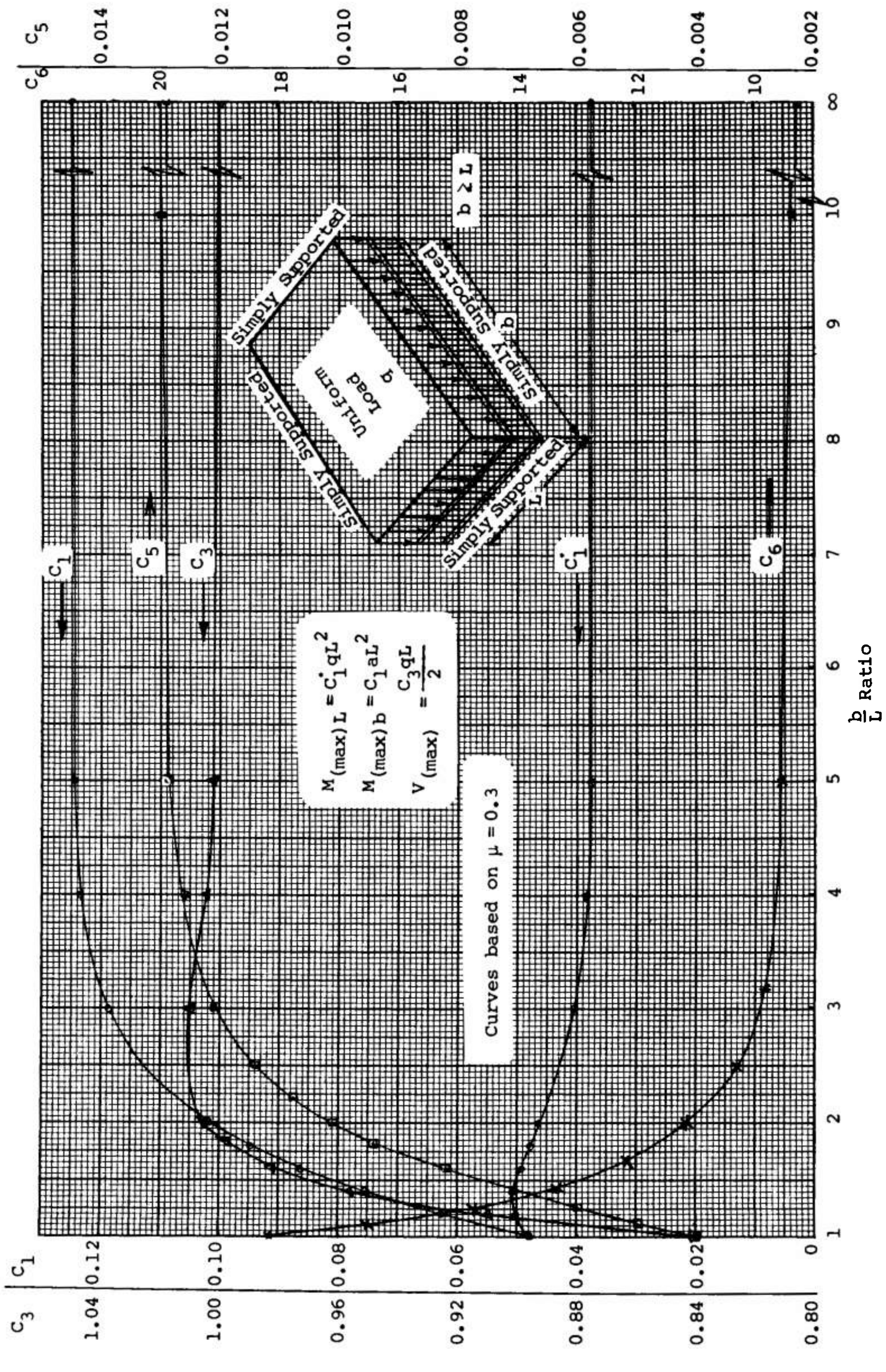


Figure 4-20. Constants for Uniformly Loaded Sandwich Panels With All Edges Simply Supported[51]

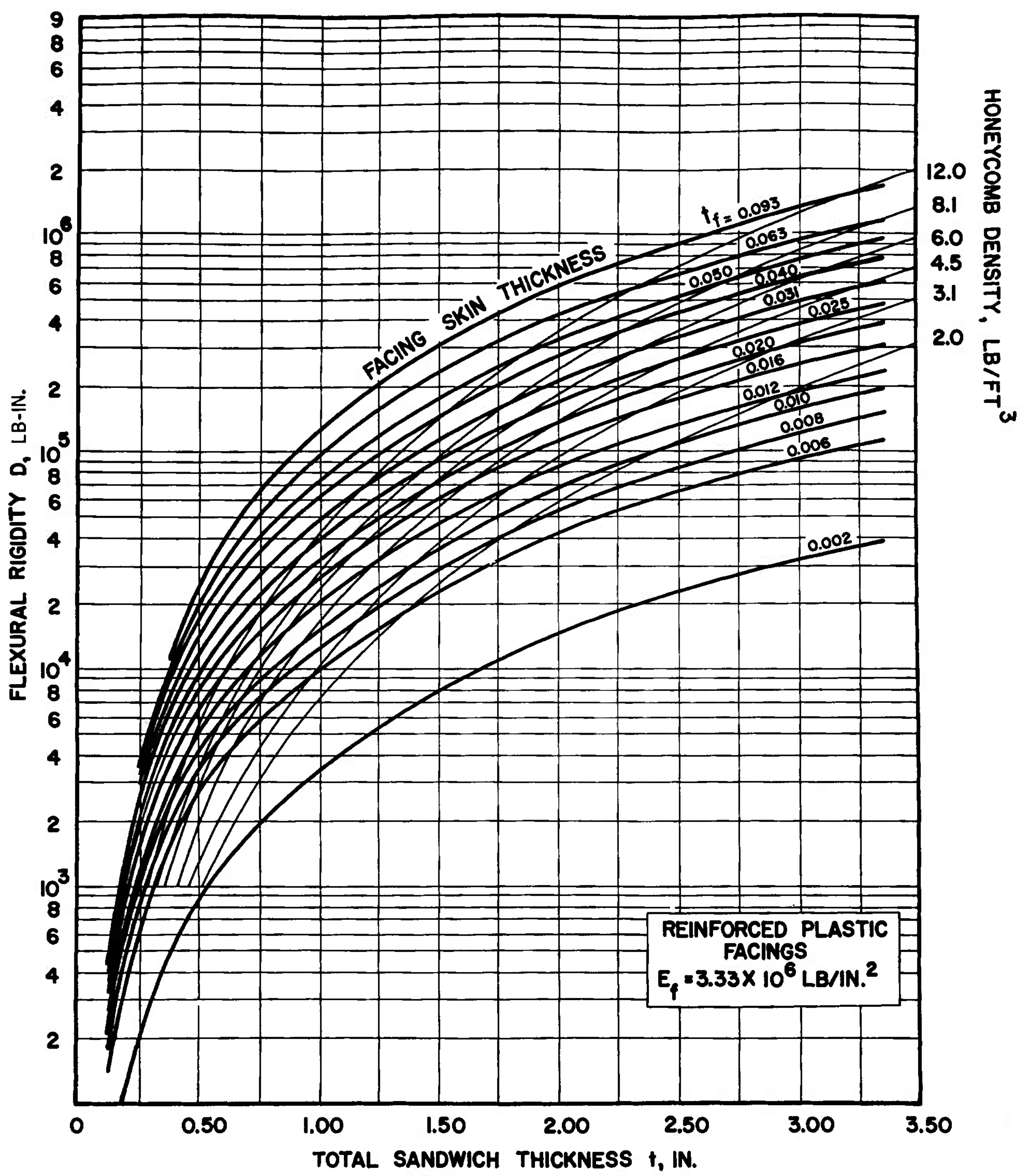


Figure 4-21. Flexural Rigidity Curve for Reinforced Plastic Facings[49] (including minimum weight curves for various honeycomb densities)

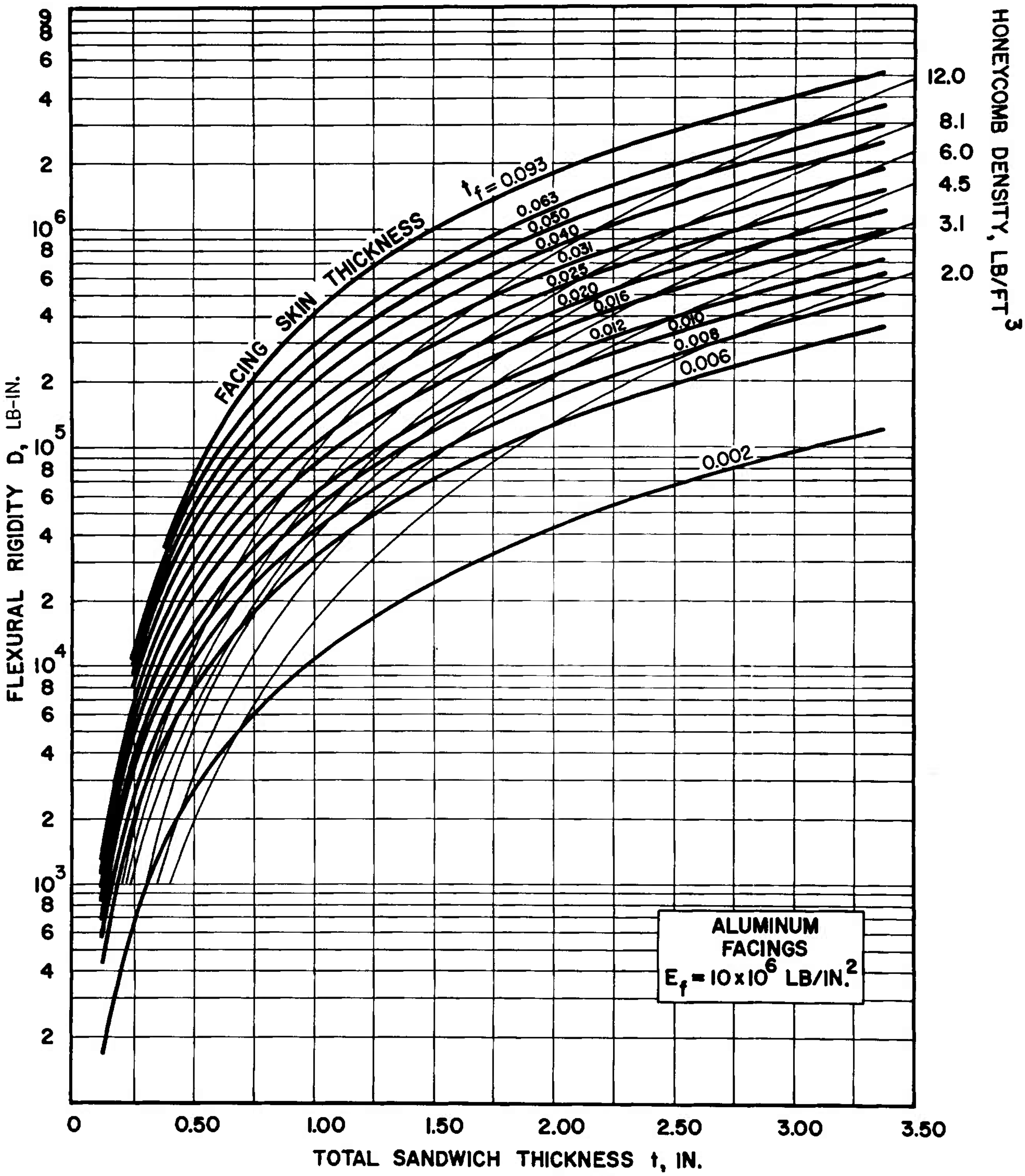


Figure 4-22. Flexural Rigidity Curve for Aluminum Facings[49] *(including minimum weight curves for various honeycomb densities)*

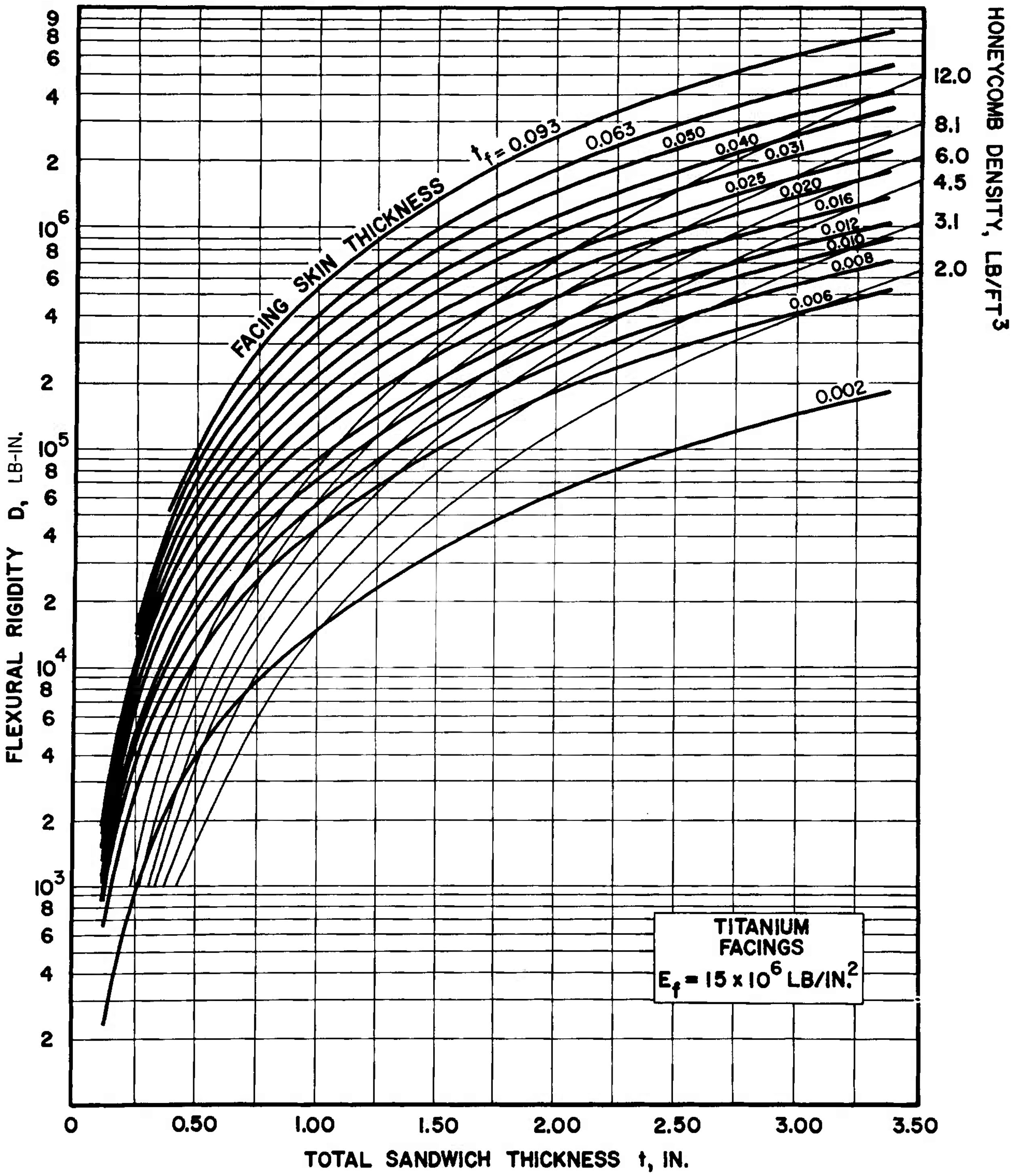


Figure 4-23. Flexural Rigidity Curve for Titanium Facings[49] *(including minimum weight curves for various honeycomb densities)*

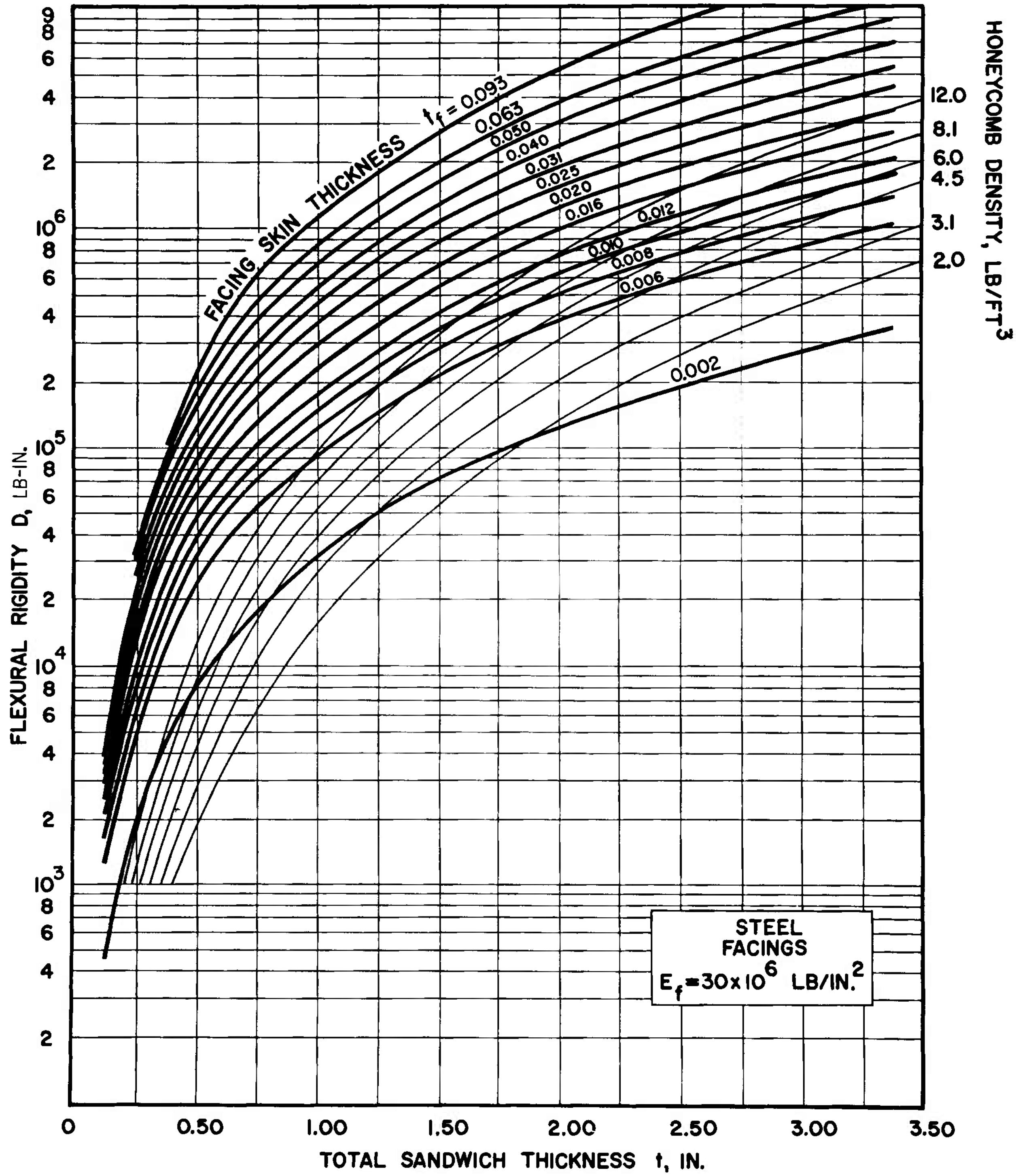


Figure 4-24. Flexural Rigidity Curve for Steel Facings[49]
(including minimum weight curves for various honeycomb densities)

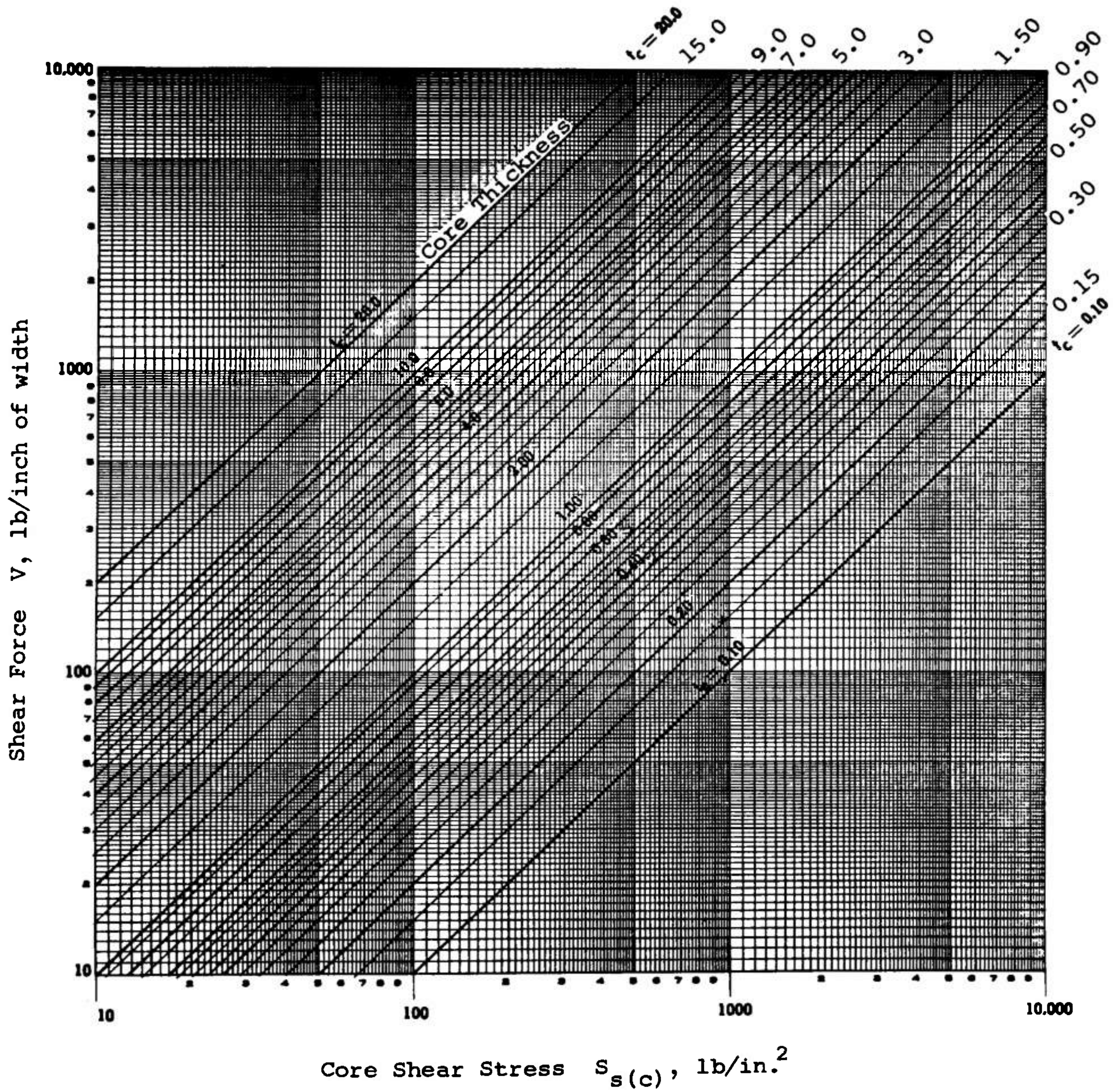
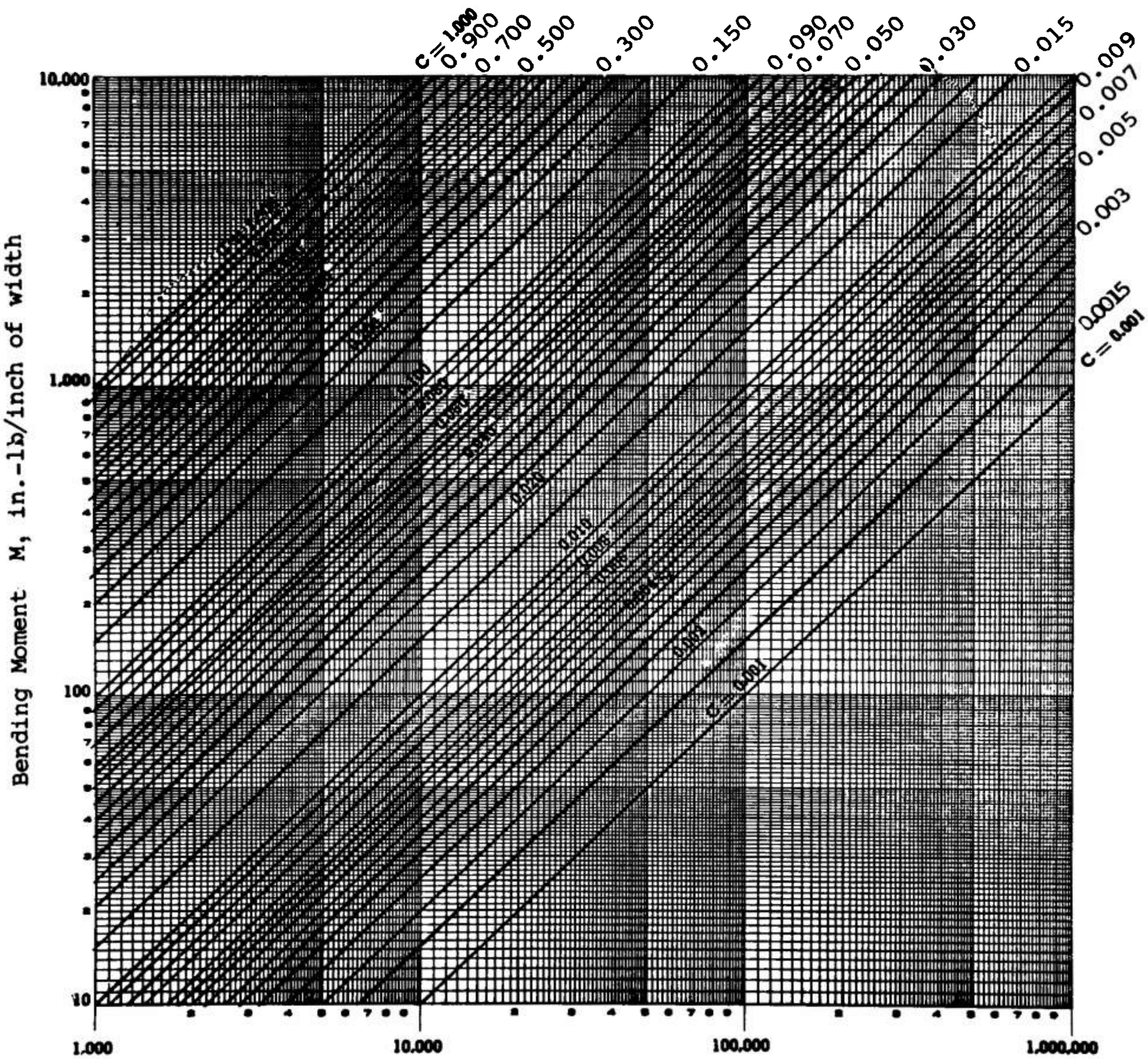


Figure 4-25. Honeycomb Shear Stress Curve[49]



NOTE: Values of C represent the minimum facing thickness required when the honeycomb core thickness is 1 in. For values of honeycomb core thickness other than 1 in., divide the C value from the curve by the actual t_c to obtain the minimum facing thickness. Any facing thickness equal to or greater than the minimum may be used.

Figure 4-26. Normal Stress Curve for Sandwich Facing Skin[49]

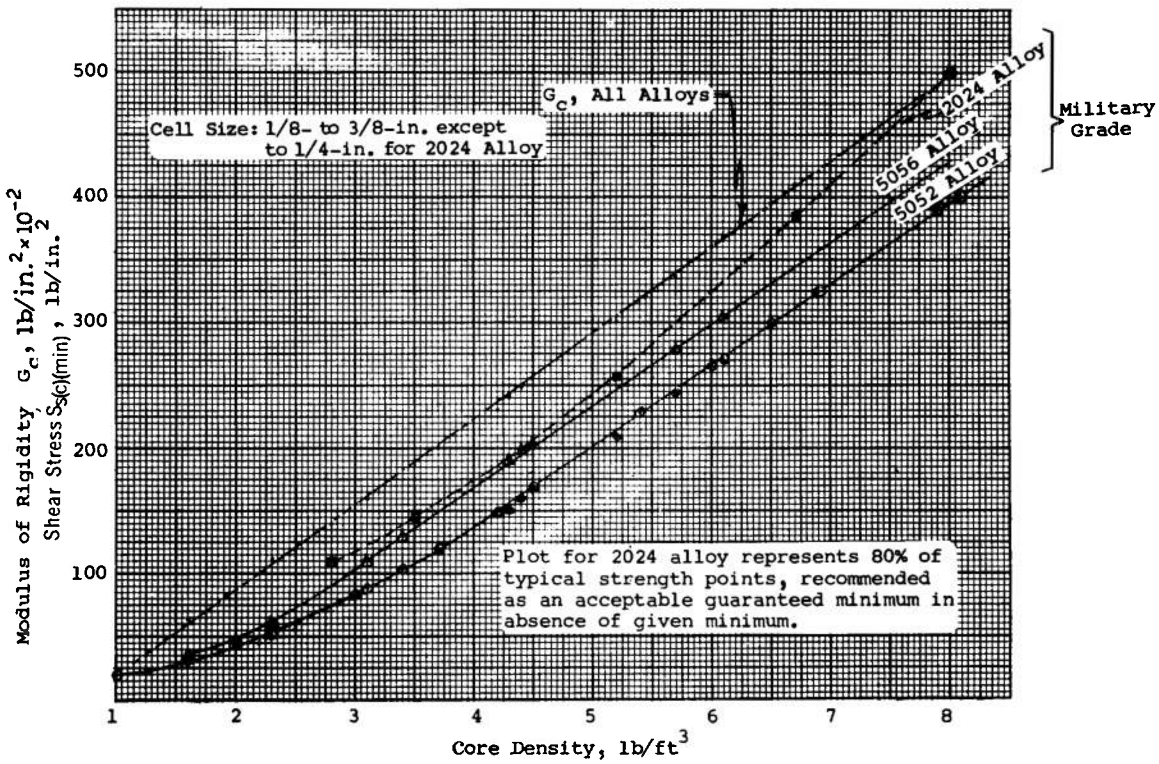


Figure 4-27. Minimum Shear Stress (W-direction) and Minimum Modulus of Rigidity for Some Aluminum Honeycomb Cores[37]

Sample Problem

It is desired to construct an 8- by 20-ft floor of a cargo vehicle using honeycomb sandwich panel. (The final floor will most likely be made of more than one panel or provided with additional supports between the longer span, generally presenting a lesser strength problem than the example.) For illustration, however, let the floor be one piece, simply supported at all edges. The cargo is assumed to be bagged flour, with an average density of 45 lb/ft^3 (Table 4-11), covering the entire floor space to a depth of 3 ft. Assume a dynamic load factor of 4.0 (par. 4-1.2).

Dynamic cargo load:

$$P = 8\,(20)\,(3)\,(45)\,(4.0) = 86{,}400 \text{ lb (total)}$$

$$q = \frac{86{,}400}{8\,(20)\,(144)} = 3.75 \text{ lb/in.}^2 \quad \text{(uniformly distributed load)}$$

Other known or set information:

$\delta_{(max)}$ = 2.5 in. (chosen to fit a normally recommended deflection range of 0.01 to 0.02 of greatest span)

$$\left.\begin{aligned} \frac{b}{L} &= \frac{240}{96} = 2.5 \\ C_1 &= 0.113 \\ C_3 &= 1.01 \\ C_5 &= 0.0114 \\ C_6 &= 10.3 \end{aligned}\right\} \text{[from Fig. 4-20]}$$

Steps of calculation are lettered and equations numbered as outlined earlier in this subparagraph.

(1) Flexural Rigidity (Eq. 4-36)

$$D = \frac{0.0114\,(3.75)\,(96)^4}{2.5} = 1.34 \times 10^6 \text{ lb-in.}$$

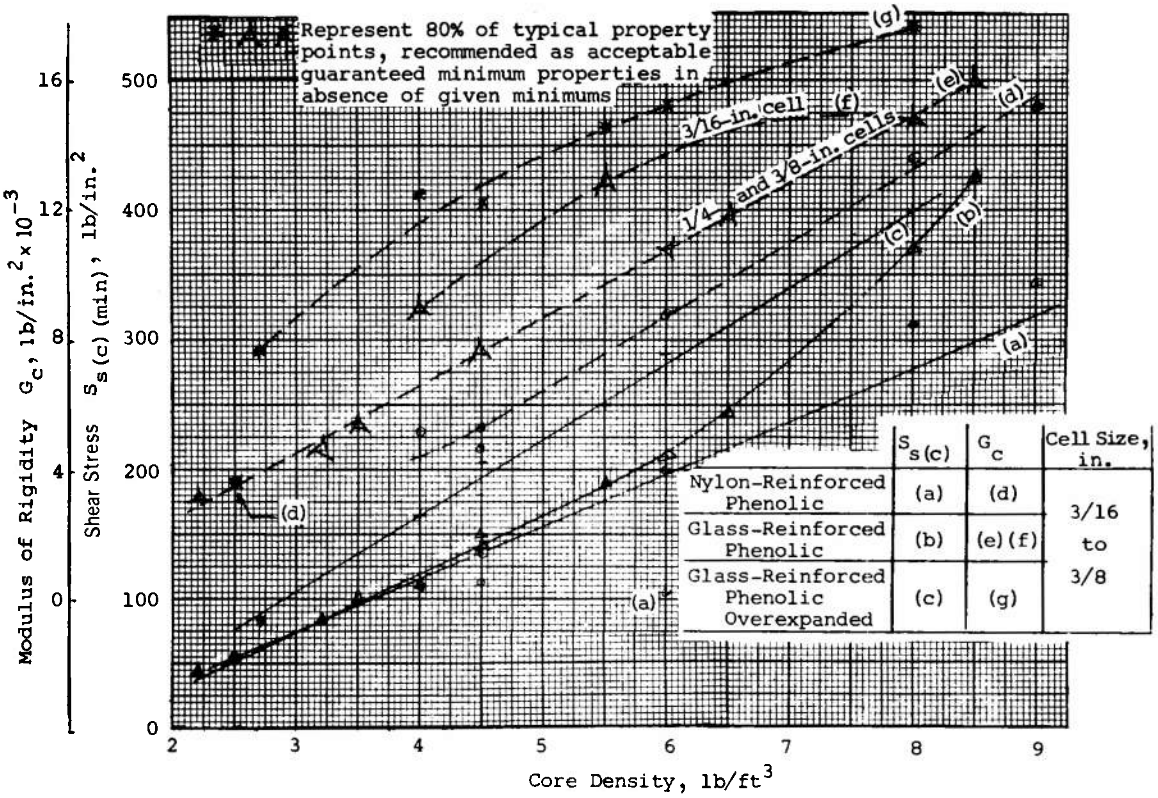


Figure 4-28. Minimum Shear Stress (W-direction) and Minimum Modulus of Rigidity for Some Reinforced Plastic Honeycomb Cores[37]

Aluminum facing skins and core will be used. Then possible facing and total thickness combinations are:

$$\left.\begin{array}{l} t_f = 0.093 \longrightarrow 0.020 \text{ in.} \\ t \cong 1.80 \longrightarrow 3.35 \text{ in.} \end{array}\right\} \quad \text{[Fig. 4-22]}$$

For impact resistance, limit $t_f \geqslant 0.025$ in.

Try, $t_f = 0.050$ in. and $t = 2.40$ in.

(2) Then, $t_c = t - 2t_f = 2.40 - 0.10 = 2.30$ in.

$$V_{(max)} = \frac{1.01\ (3.75)\ (96)}{2} = 182 \text{ lb/in.} \quad \text{[Fig. 4-20]}$$

therefore,

$$S_{s(c)} = 80 \text{ lb/in.}^2 \qquad \text{[Fig. 4-25]}$$

(3) From Fig. 4-27,

$$\left.\begin{array}{l} S_{s(c)} = 103 \text{ lb/in.}^2 \text{ (5056 alloy)} \\ \quad\;\; = 83 \text{ lb/in.}^2 \text{ (5052 alloy)} \end{array}\right\}$$

for core density 3 lb/ft^3, consider this density as minimum for convenience

Preliminary margins of safety core,

$$MS = \frac{103}{80} - 1 = 0.29 \text{ (5056)}$$

$$= \frac{83}{80} - 1 = 0.04 \text{ (5052).}$$

For higher core density, $S_{s(c)(min)}$ is substantially higher (Fig. 4-27). At this point density has not been established, but first choice of t_f and t correspond to much higher densities than needed.

(4) Use 7075-T6 clad aluminum for facing skin,

$$S_{yield} = 67{,}000 \text{ lb/in.}^2 \quad \text{[Table 4-6]}$$

$$M_{(max)b} = 0.113\ (3.75)\ (96)^2 = 3900 \text{ in.-lb/in.} \quad \text{[Fig. 4-20]}$$

$$C = 2.30\ (0.050) = 0.115 \text{ in.}^2 \quad \text{[Eq. 4-38]}$$

Then, $S_f \cong 35{,}000$ lb/in.^{2} [Fig. 4-26]

Preliminary margin of safety for facing,

$$MS = \frac{S_{yield}}{S_f} - 1 = \frac{67}{35} - 1 = 0.91$$

(5) Weight Optimization: Use density = 4.5 lb/ft^3, which represents approximately the lowest usable density for D and $t_{f(min)}$ determined in step (1).

(6) New thicknesses

$t_f = 0.025$ in., $t = 3.35$ in. [Fig. 4-22]

then $t_c = 3.30$ in.

Recycle through steps (2) to (4)

$S'_{s(c)} \cong 55$ lb/in.^{2} $\qquad S'_{s(c)(min)} = 168$ lb/in.^{2}

Core factor of safety, using 5052 alloy

$$SF = \frac{168}{55} = 3.06$$

Final core cell size and strip (foil) thickness t_s can be made from manufacturer's tabulations of core data in terms of density (e.g., Ref. 38).

$$C' = 3.30\ (0.025) = 0.0825 \text{ in.}^2$$

$$S'_f = 48{,}000 \text{ lb/in.}^2$$

Facing skin margin of safety

$$MS = \frac{67}{48} - 1 = 0.39$$

Check on actual maximum deflection (G_c from Fig. 4-27)

$$\delta_{(max)} = \frac{0.0114\ (3.75)\ (96)^4}{1.45 \times 10^6} + \frac{0.0114\ (10.3)\ (3.75)\ (96)^2}{3.30\ (2.58 \times 10^3)} \quad \text{[Eq. 4-32]}$$

$$= 2.51 + 0.048 = 2.56 \text{ in. (0.0107 of span)}$$

Resultant weight of floor exclusive of adhesive or inserts. Weights of facing materials are given in Table 4-6.

core $= 4.5\left(\frac{3.30}{12}\right) = 1.24$ lb/ft^2

facing skins $= 2(0.0145)\left(\frac{0.025}{0.001}\right) = 0.73$ lb/ft^2

floor $= 1.97$ lb/ft^2

total floor $= 1.97\ (8)\ (20) = 315$ lb

The designer should investigate other areas of optimization, possibly the use of different core and facing skin materials. The use of unequal facing skins (thicker on top) for additional impact provision is worth consideration in problems of this type.

4-2.3.3 Stressed-skin Structures

Beams or versions of beams are the primary load-carrying members in any construction. The stressed-skin structure simulates a beam by combining a skin (or panel) with a minimum of stiffeners into a high strength-to-weight ratio unit. Analogy of the stressed-skin structure to an I-beam is quite evident, as shown in Fig. 4-29. Reinforcements all along two edges of the structure approximate flanges of the I-beam while the skin and its lesser reinforcements correspond to the web[52]. The web analogy is used only as an introductory illustration, since unlike the shear resistant web of an integral beam, the thin stressed skin is characterized by considerable local buckling between supports. Design of the stressed-skin structure considered here permits such buckling without failure of the structure as a whole and makes this construction unique. The skin referred to here can be any one of several types such as the simple sheet, multiple sheets, or various corrugated sheets (Fig. 4-29). It is to be emphasized, however, that the skin is a real, load-bearing, structural member, not merely a covering-over framing. Much of the theory and development of the stressed-skin technique stems from aircraft structures, but weight reduction made possible thereby is no less important in automotive structures that can take advantage of it. Sandwich panels, also exhibiting stressed "skins", are treated separately in par. 4-2.3.2 and plate "skins" of heavier vehicles are discussed in Sections II, III, and IV of this chapter.

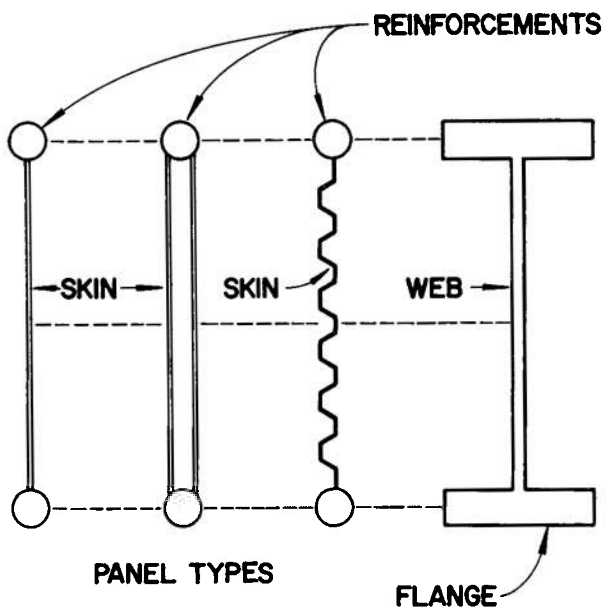


Figure 4-29. Analogy of Stressed-skin Structure to an I-beam

4-2.3.3.1 Distribution of Load

The components of stressed-skin structures have different modes of strength that can be combined successfully and used only if they act together as a unit. This calls for proper load distribution among them.

A portion of a simplified stressed-skin structure is shown in Fig. 4-30(A). When loaded as a beam (Fig. 4-30(B)), members such as 1 and 2 are put under tension and compression, respectively, because of bending while members 3 are under axial compression. The skin ties all members together, strengthening the unit through its tensile property and absorbs the shearing loads developed.

Load distributing action of the skin is further illustrated by considering a single "block" (*abcd*) of the beam, as shown in Fig. 4-30(B). Behavior of the skin can then be compared to the action of a pair of wire-like members replacing the skin and forming diagonals $\overline{ac}$ and $\overline{bd}$ of the "block"[52]. Under no load, the "wires" have equal tension, but when load is applied, deflection and shear cause increase of tension in $\overline{bd}$ and decrease of tension (relative compression) in $\overline{ac}$ is shown exaggerated in Fig. 4-30(C). Tension can be resisted; compression represents unstable loading on the skin which rapidly results in a series of ripples, with crests and depressions of buckled skin running parallel to the direction of diagonal tension developed (Fig. 4-31). The magnitude of ripples increases sharply as the load is increased. In general, direction of ripples is not parallel to diagonals of panels; but such buckling does not indicate failure of a stressed-skin structure if overall unit integrity is maintained by supports and if resulting local deformations are permissible. The extent of buckling allowed in the skin is such that full working stress of the material is developed. However, if distribution and absorption of tensile and shearing loads are to be realized, the skin must be well joined to all supports.

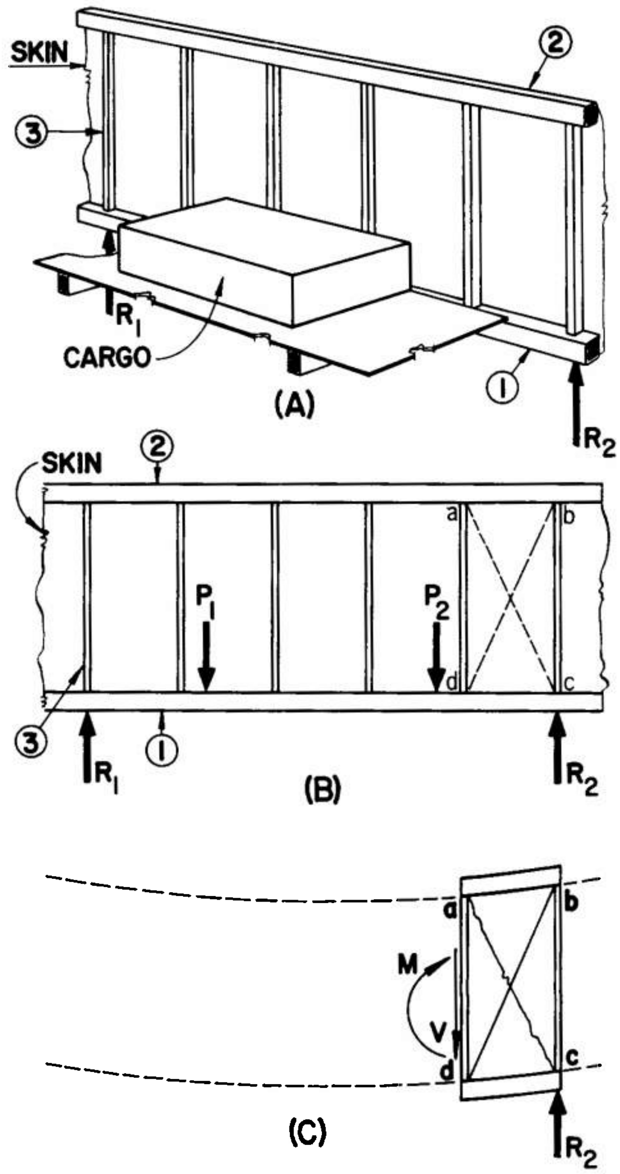


Figure 4-30. Simplified Stressed-skin Structure

In the foregoing description, external loading on panel $\overline{abcd}$ (Fig. 4-30(C)) and the beam (Fig. 4-31) was upward; with downward loading, the roles of "wires" $\overline{ac}$, $\overline{bd}$, and orientation of the buckling pattern are reversed. Load applications in other directions relative to a typical panel produce effects similary to either the upward or downward loading case.

Actual conditions existing in a stressed-skin structure are, of course, much more complex than the approximation presented; but do acquaint the reader with the problem. Full solutions to some problems of this technique are lacking, but work in this area continues[53]. A statement of the existing engineering theories is in order, however, since they relate to load distribution.

Theories assume the skin of the beam in Figs. 4-30 and 4-31 to be in a pure shear condition for applied loads up to the skin buckling load (par. 4-2.3.3.2). However, after buckling or rippling, theories differ as to how the supported skin carries further shear load. The complete diagonal tension theory assumes that all the shear resistance of the skin after initial buckling is due to increased diagonal tension on planes parallel to the ripples, while the diagonal compressive stress (perpendicular to ripples) which caused buckling remains at that original value[54]. The incomplete diagonal tension theory[55] proposed that post-buckling shear strength of the skin is due to a combination of the pure shear and the complete diagonal tension conditions. The extent of approach to the complete tension case is expressed by a coefficient and varies from structure to structure. The latter theory is more indicative of actual stress conditions in the skin, but yields slightly higher margins of safety[54] than obtained by the former theory. Ref. 53 deems the method easy to apply and is well substantiated by experimental evidence in other analyses in existence that depend on more advanced methods. The Ref. 55 method is used for analysis in this handbook.

4-2.3.3.2 Types of Failure

Failure modes of stressed-skin structures are evident from their loading described in the previous subparagraph. Possible types of failure are: (a) column failure of flange-like members (components 1 or 2 in Fig. 4-30(B), depending on load direction); (b) tensile failure of flange members (the opposite flange to that in (a)); (c) column failure of skin stiffeners (components 3, Fig. 4-30(B)). Two distinct types of failure are recognized in stiffeners—(1) general column

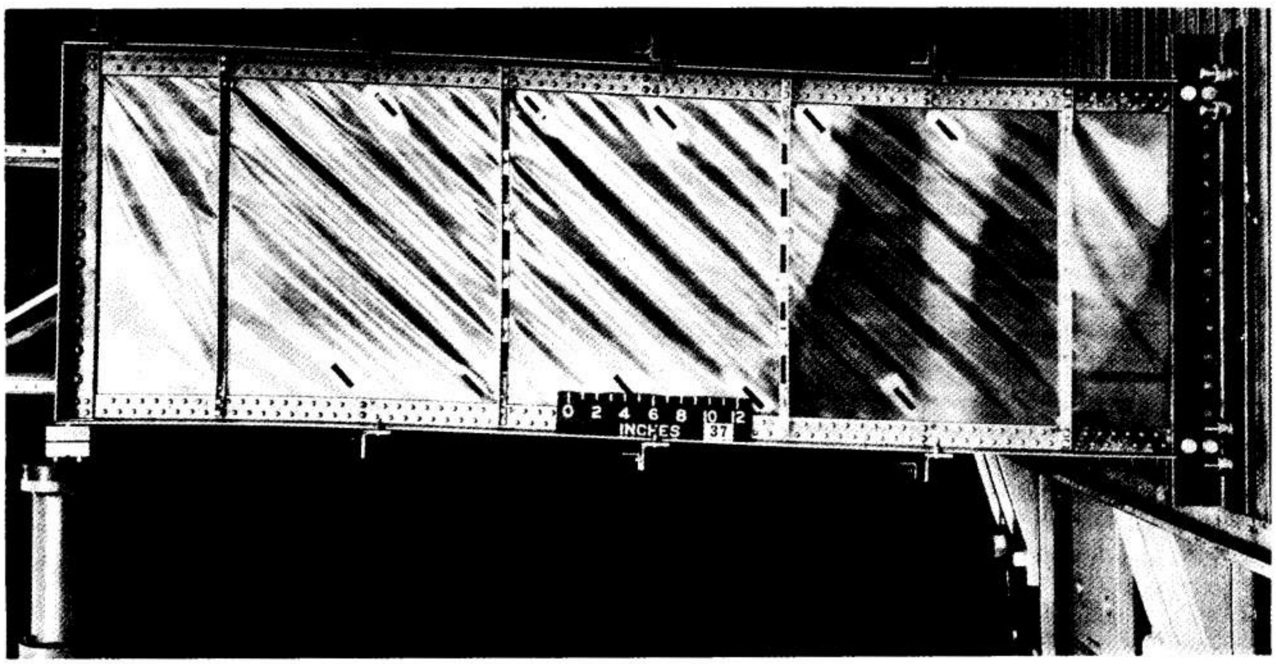


Figure 4-31. Diagonal Tension Beam

failure by instability of the member, occurring without prior bowing, and (2) forced-crippling failure (local failure) in which the buckled webs are "forced" into the stiffeners. An illustration of this failure type appears in Fig. 4-32. Additional types of possible failure are (d) tensile (tear) failure of the skin, and (e) shear failure of the skin due to excessive buckling. Modes (b) and (d) are less likely to occur because of inherent material tensile properties, but are possible under impacts or penetrations to which vehicle structures are exposed in a military environment. Therefore, emphasis is placed on failure modes (a), (c), and (e).

A general note about the limitations of the working equations to be presented is in order. The Ref. 55 method is based on a large number of tests performed on highly loaded skin-web beams. Web materials used were two aluminum alloys, 2024-T3 and Alclad 7075-T6 (formerly 24S and 75S alloys), and were tested under a pure shear loading. Much work was also done on curved webs under torsion, but this is not included here. A considerable amount of empirical data is part of the method, and caution is required if their numerical range is exceeded in a specific problem encountered by the designer.

The beam under consideration here has large depth, small flange width, and a

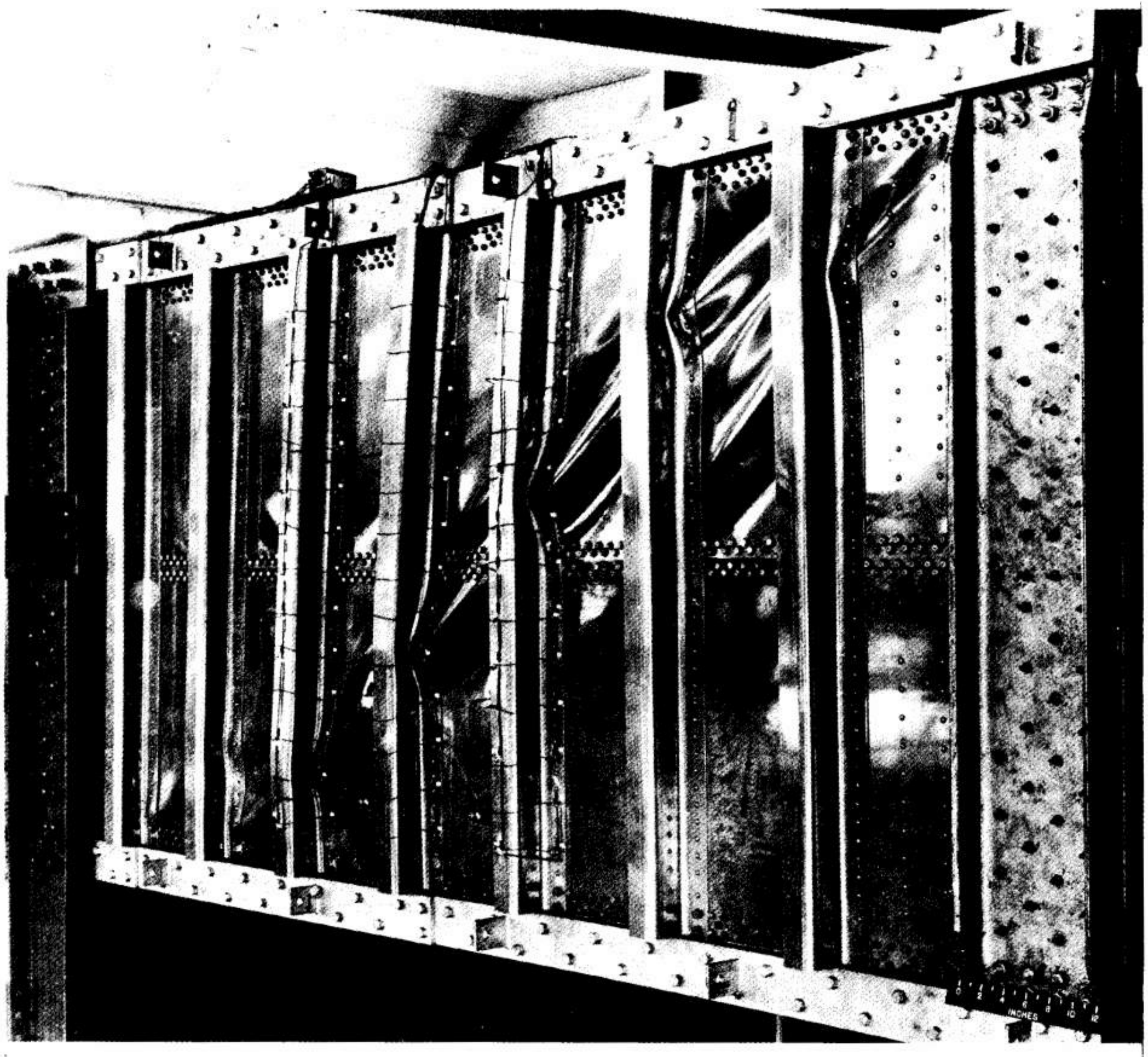


Figure 4-32. Failure by Forced Crippling of Uprights on a Large Skin-web Beam

characteristically thin skin-web (test beams were stabilized against lateral buckling of flanges and torsion). This geometry makes possible some simplifying assumptions[54,55]: (a) only the flanges are subject to the tensile and compressive stresses resulting from the bending moment; (b) variations of shear stresses are small over the depth of the skin and thus, are assumed constant; and (c) the skin-web is subjected to pure shear stress during beam loading up to the point of skin buckling, and its governing buckling strength is determined by shear.

Strength of the Skin

The critical shear stress (buckling stress) in a flat isotropic sheet, formed into panels of height h and width b by supports, as in Figs. 4-30 and 4-31, is given in the following forms in numerous sources for various theoretical restraints[56-58]:

$$S_{s(cre)} = K_s \left[\frac{\pi^2 E}{12(1-\mu^2)}\right]\left(\frac{t}{b}\right)^2 = K_s' E\left(\frac{t}{b}\right)^2 \quad (4\text{-}40)$$

$S_{s(cre)}$ = critical elastic (buckling) stress due to shear loading on skin, lb/in.^{2}

K_s, K_s' = theoretical buckling coefficients for shear loading (functions of sheet dimension ratio h/b and type of edge restraint), dimensionless

$$K_s' = K_s \left[\frac{\pi^2}{12(1-\mu^2)}\right]$$

E = modulus of elasticity of material, lb/in.^{2}

μ = Poisson's ratio for the material, dimensionless ($\mu \cong 0.3$ for metals. See Table 4-3 for specific materials.)

t = sheet thickness, in.

b = shorter dimension of sheet (usually width), in.

Eq. 4-40 is not easily used; some disagreement exists between values of buckling coefficients appearing in the literature as reported in Refs. 55 and 59. Moreover, small differences in assumptions made and the existence of modified forms (for instance K_s and K_s' in Eq. 4-40) often contribute to the problem. Also, adequate formulations for actual edge restraints—in general, neither freely supported nor fixed—are not yet available. Therefore, the Ref. 55 method uses a semi-empirical formula similar in form to Eq. 4-40 to interpolate between the practical shortcomings of listed buckling coefficients.

$$S_{s(cre)} = K_{ss} E\left(\frac{t}{b}\right)^2 \left[R_h + 0.5(R_b - R_h)\left(\frac{b}{h}\right)^3\right] \text{lb/in.}^2 \; (b \leqslant h) \quad (4\text{-}41)$$

where

K_{ss} = theoretical shear buckling coefficient for a sheet of height h and width b, with all edges simply supported (Fig. 4-33), dimensionless

b = actual unsupported width of skin (Fig. 4-33), in. (for riveted skin, distance between consecutive stiffener rivet centroids see Case 1; for welded skin, distance from end to beginning of adjacent stiffeners approximated by Case 2.)

h = actual unsupported height of skin (Fig. 4-33), in. (similar explanatory note for distance between flanges applies here as for distance between stiffeners given with dimension b.)

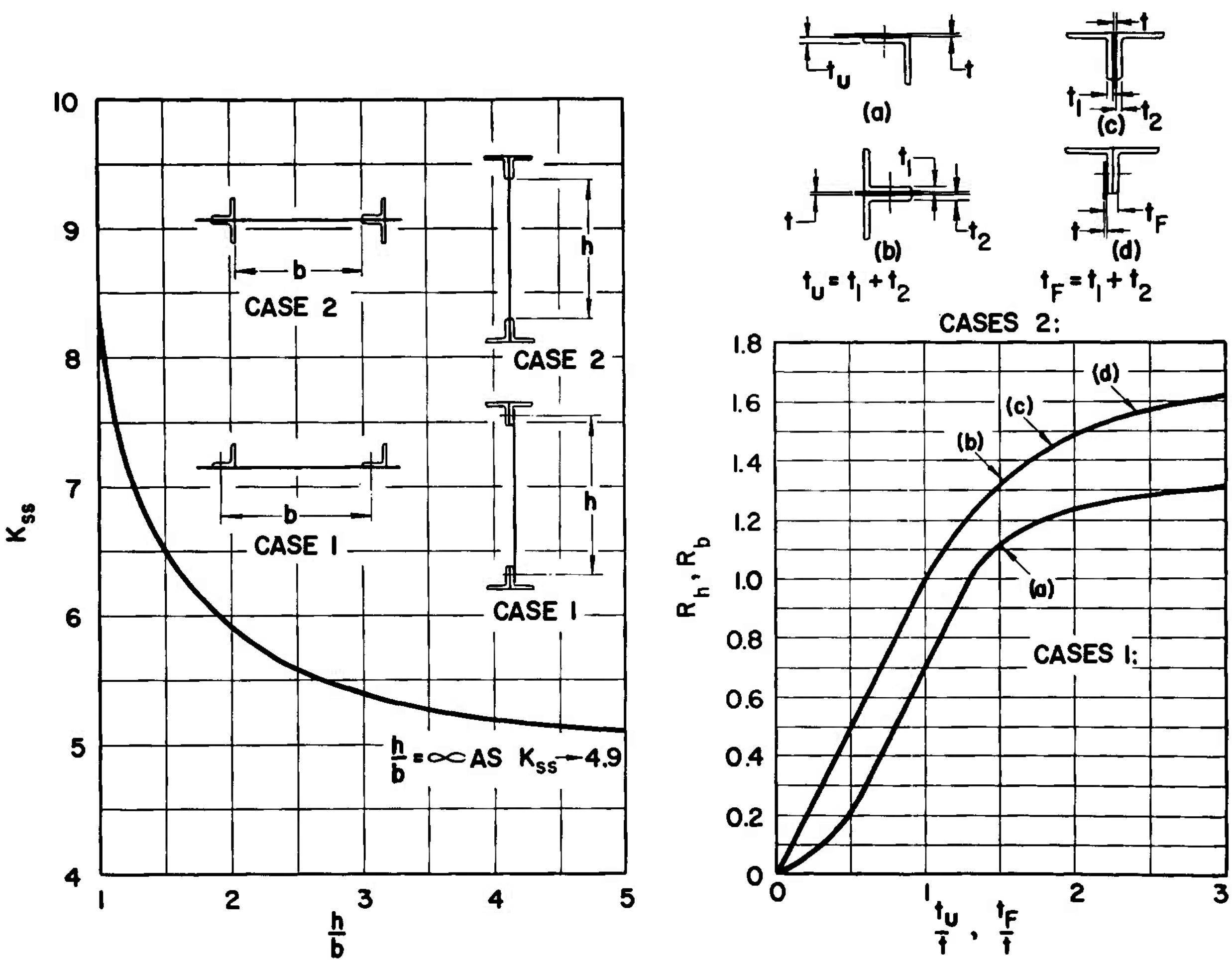
R_b, R_h = empirical edge restraint coefficients (skin of flange and skin to stiffener, respectively, see Fig. 4-33, dimensionless. Use of Case 1 or 2 is given under dimension b.)

$$R_b, R_h = F\left(\frac{t_u}{t}, \frac{t_F}{t}\right)$$

t_u = thickness of stiffener in contact with skin, in. (Fig. 4-33).

t_F = thickness of flange in contract with skin, in. (Fig. 4-33).

t = skin thickness, in.



(A) THEORETICAL COEFFICIENTS FOR PLATES WITH SIMPLY SUPPORTED EDGES

(B) EMPIRICAL EDGE RESTRAINT COEFFICIENT

DATA BASED ON BOLTED OR RIVETED ATTACHMENT OF SKIN TO SUPPORTS IN ALL CASES; DISTINCTION IS MADE BETWEEN SUPPORTS ON ONE OR BOTH SIDES OF SKIN. THE LATTER CONSTRUCTION IS NOT COMMON IN AUTOMOTIVE STRUCTURES, BUT CAN BE USED TO APPROXIMATE THE CONDITION OF SKIN WELDED TO SUPPORTS.

Figure 4-33. Shear Buckling Coefficients[55]

Eq. 4-41 applies only for panel dimensions $b \leqslant h$; however, substitution of h for b, b for h, R_h for R_b, and R_b for R_h makes that equation usable for the range $b > h$. A similar transformation is necessary when using Fig. 4-33. Eqs. 4-40 and 4-41 are valid only for critical stress values less than the yield stress of the material in shear.

In the inelastic or plastic range, strain increases faster than stress; therefore actual shear stress in this region will be less than that predicted by equations for the elastic range. Relations between the two regions depend on type of material for a given loading, and are normally obtained by stress-strain tests on specific materials. Fig. 4-34 gives the relation between $S_{s(cre)}$ (elastic shear buckling stress) and $S_{s(cr)}$ (corresponding inelastic stress) for two particular skin materials, showing that, in the elastic range, $S_{s(cre)} = S_{s(cr)}$. Values of $S_{s(cre)}$ obtained from Eq. 4-41 are then to be referred to Fig. 4-34 (or similar curves of the material under investigation) for plasticity correction; i.e., if $S_{s(cr)} < S_{s(cre)}$ in a given case, the value of $S_{s(cr)}$ should be used for further design.

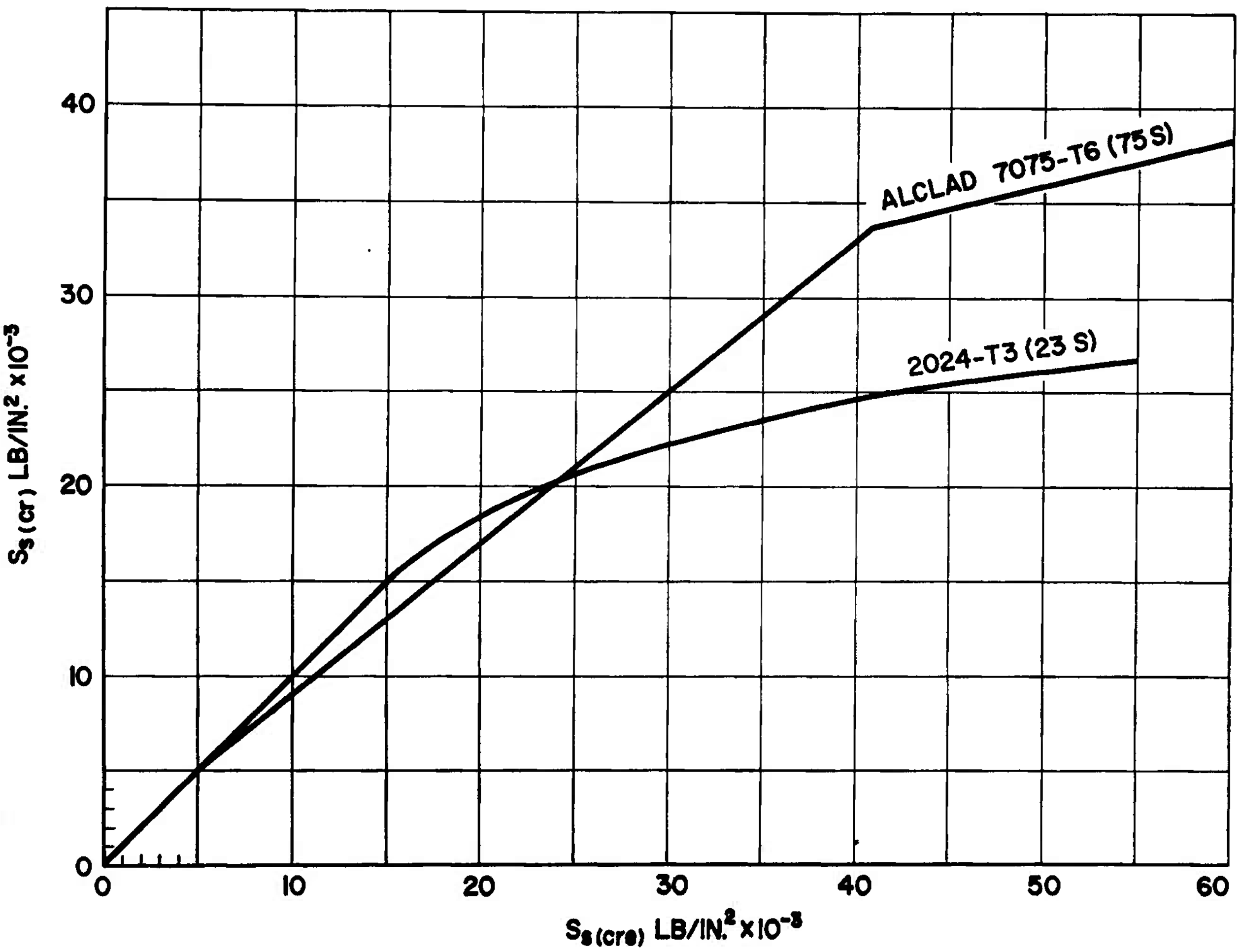


Figure 4-34. Plasticity Correction[55]

After local buckling, the incomplete diagonal tension theory represents the stress condition in the web as

$$S_s = S_{s1} + S_{s2} = (1 - n)\,S_s + nS_s \qquad S_s \geqslant S_{s(cr)} \tag{4-42}$$

where

S_s = total shear stress inweb, lb/in.^{2}

S_{s1} = portion of shear stress inweb resisted by pure shear, lb/in.^{2}

S_{s2} = portion of shear stress in web resisted by diagonal tension on appropriate planes, lb/in.^{2}

n = diagonal tension factor, dimensionless

The value of total shear stress S_s is computed by treating it as if it were a nominal or average shear stress in the skin-web. For beam cross sections described earlier, this relation is

$$S_s = \frac{V}{Ht}, \ \text{lb/in.}^2 \qquad S_s \geqslant S_{s(cr)} \tag{4-43}$$

where

V = vertical shear load applied on web, lb

H = distance between centroids of flanges, in.

For other beam sections, shear stress in terms of applied load takes on more complicated forms[54,55,60]. The diagonal tension factor n was determined by a large number of experiments on beams[55]:

$$\left.\begin{aligned} n &= \tanh\left[\left(0.5 \log_{10} \frac{S_s}{S_{s(cr)}}\right)\right], \quad S_s \geqslant S_{s(cr)} \\ n &\approx 0.434\left(\rho + \frac{\rho^3}{3}\right) \qquad 1 < \frac{S_s}{S_{s(cr)}} < 2 \end{aligned}\right\} \tag{4-44}$$

where

$$\rho = \frac{S_s - S_{s(cr)}}{S_s + S_{s(cr)}}\text{, dimensionless}$$

The first of Eqs. 4-44 is cumbersome; therefore, it is plotted in Fig. 4-35 as a function of the loading ratio $S_s/S_{s(cr)}$ and may be readily used after $S_s/S_{s(cr)}$ is determined by Eqs. 4-43 and 4-41, respectively. Limits of incomplete diagonal tension occur when $S_s/S_{s(cr)}$ approaches values of 1.0 and ∞; the corresponding values of n then are 0 and 1.0 as is shown in Fig. 4-35. Physically, $n = 0$ occurs often in thick, nonbuckling webs of beams used in ordinary structures, where $S_s/S_{s(cr)} < 1$; $n = 1$ is approximated only in very highly loaded skin webs, where $S_s/S_{s(cr)}$ is very large. The peak value of the nominal shear stress in the web is related empirically to the shear stress of Eqs. 4-42 or 4-43 by use of two factors to correct elementary theory—angle factor C_1 and stress concentration factor C_2. C_1 corrects for variation of the angle (theoretically 45°) which resulting tensile forces in the skin make with shear forces applied parallel to the edges. Practical values of tension angles are less than theoretical—40° being a good approximation. Ref. 55 assumes C_1 to be a function of n^2 (diagonal tension factor). C_1 is very small for tension angles near 45° (Fig. 4-36), but becomes more significant with decreasing angle, as is the case for curved webs (also treated in Ref. 55). The stress concentration factor C_2 acts for increased stress at skin panel corners. Diagonal tension developed in the skin exerts a pull on flange members causing them to deflect between skin stiffeners. This results in stress concentrations around panel corners where flanges are restrained by the column action of stiffeners. C_2 is assumed to be a linear function of n; it is also a function of a flange-flexibility parameter d which must be used in conjunction with Fig. 4-37. Parameter d can be approximated by Eq. 4-45 for tension angles "slightly less than 45°" (Ref. 55).

$$d = 0.7\, b_1 \left[\frac{t}{H(I_{TF} + I_{CF})}\right]^{1/4}\text{, dimensionless} \tag{4-45}$$

where

b_1 = skin stiffener spacing or pitch, in.

I_{TF}, I_{CF} = moments of inertia of tensile and compressive flange cross sections, respectively, about their neutral axes, in.^{4}

Therefore, the maximum web shear stress is

$$S_{s(max)} = S_s\,(1 + n^2 C_1)\,(1 + nC_2)\text{, lb/in.}^2 \tag{4-46}$$

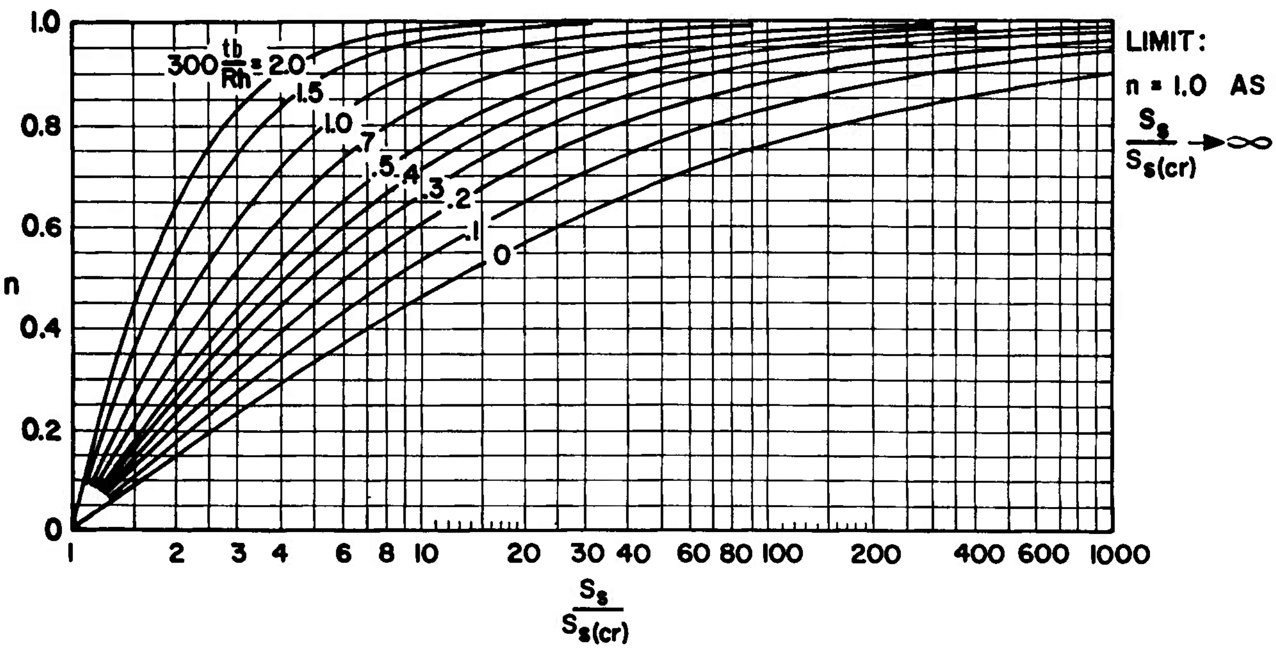


Figure 4-35. Diagonal Tension Factor n (Ref. 55)

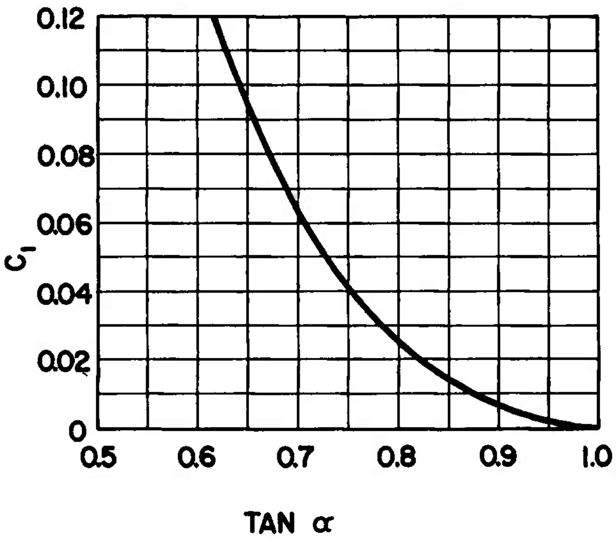


Figure 4-36. Angle Factor C_1 (Ref. 55)

where

C_1 = $(1/\sin 2a) - 1$, angle factor, dimensionless [Fig. 4-36]

a = angle of diagonal tension measured from beam longitudinal neutral axis, deg. a is obtained from Fig. 4-38 after ratio $S_{c(u)}/S_s$ is determined from Fig. 4-40.

C_2 = stress concentration factor, dimensionless [Fig. 4-37]

Allowable web shear stresses for the two materials were also obtained empirically in tests that produced failure in webs at points of attachment to flanges; they appear in Fig. 4-39 and represent "the line 10 percent below the average of the scatter band" of failure. Approximately 85 percent of the beams were within ±5 percent, and nearly all beams were within ±10 percent of the average stress at failure[55]. Results of Eq. 4-46, used in conjunction with Fig. 4-39, serve to establish a margin of safety for the skin. Allowable stresses of Fig. 4-39 apply strictly for webs attached by means of tightly snugged bolts and with heavy washers for bearing, or for webs clamped between supports on each side. However, supplementary tests indicated that the curves given may be used for other types of attachment with the following reservations:

a. When bolts are used without washers to increase bearing area, decrease the value given for $S_{s(a)}$ by 10 percent.

b. When rivets are used for attachment, increase the value given for $S_{s(a)}$ by 10 percent. This does not apply to countersunk head rivets. These require a decreased allowable stress.

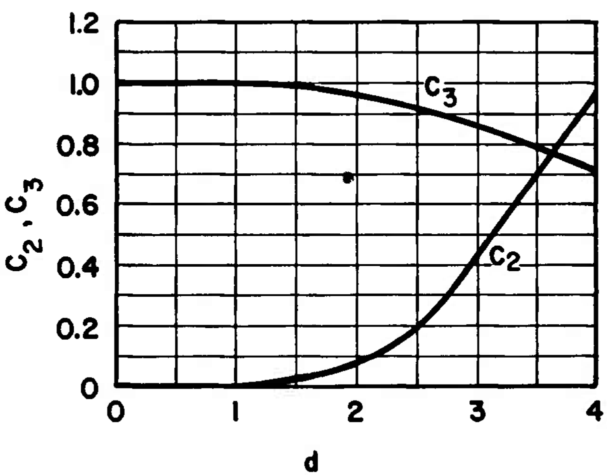
c. If rivets are assumed to loosen in service, use $S_{s(a)}$ as given.

Webs supported by flanges and uprights on one side only, and having $h/t < 100$ also require a reduction of allowable web stress as given by Fig. 4-39. Furthermore, a reduction of allowable web stress is also warranted at locations where concentrated loads are applied. Because of stress concentration, shear stress cannot be assumed uniform over the web. Local increase of web thickness or other reinforcement is often required.

Strength of Skin Stiffeners

Average compressive stress in skin stiffeners or uprights is

$$S_{c(u)} = \frac{nS_s \tan a}{\dfrac{A_{ue}}{b_1 t} + \dfrac{b_e}{b}} = \frac{nS_s \tan a}{\dfrac{A_{ue}}{b_1 t} + 0.5\,(1-n)}, \text{ lb/in.}^2 \quad (4\text{-}47)$$



$$d = 0.7 b_1 \left[\frac{t}{H(I_{TF} + I_{CF})} \right]^{\frac{1}{4}}$$

Figure 4-37. Stress Concentration Factors C_2 and C_3 (Ref. 55)

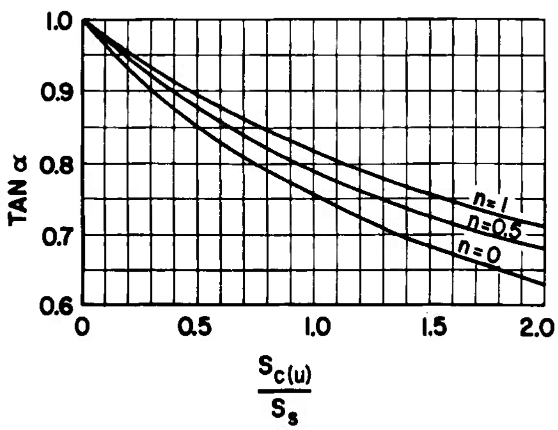


Figure 4-38. Incomplete Diagonal Tension Angle[55]

where

A_{ue} = effective cross-sectional area of stiffener exclusive of skin itself, in.^{2}

$A_{ue} = \dfrac{A_u}{1 + \left(\dfrac{e}{r}\right)^2}$ for skin stiffener on one side of skin only, or for stiffeners of unequal section on opposite sides of the skin

$A_{ue} = A_u$ for stiffeners of equal section on both sides of the skin ($e = 0$)

A_u = ordinary cross-sectional area of stiffener exclusive of skin itself, in.^{2}

e = eccentricity (distance from centroid of A_u to midthickness of skin), in.

r = radius of gyration of A_u centroid about an axis parallel to the skin, in.

b_e = 0.5 b_1 (1 - n), effective width of skin resisting compression together with stiffener even after initial buckling (empirically determined), in.

n, S_s, a, b_1, and t are defined for Eqs. 4-41 to 4-46.

$S_{c(u)}$, as determined by Eq. 4-47, is an average of the compressive stresses along the length of the stiffener. The actual stress varies from a maximum—$S_{c(u)(max)}$, Eq. 4-48—at the midlength of the stiffener to a minimum at its two ends. For stiffeners that are symmetrical about the skin-web—i.e., stiffeners of equal cross section placed on opposite sides of the web—the stress is considered uniform over the cross section. For asymmetrically arranged stiffeners—i.e., single stiffeners on one side of the web, only—the stress varies across the stiffener cross section from a maximum at the surface adjacent to the web to a minimum at the surface away from the skin. In these cases, $S_{c(u)}$ is an average of these stresses. (See additional discussion on "Stiffeners on One Side of Web Only", following Eq. 4-53.) $S_{c(u)}$ is more conveniently determined by means of Fig. 4-40 instead of Eq. 4-47.

The maximum value of the compressive stress in the upright $S_{c(u)(max)}$ is deemed a better criterion for establishing allowable stress in this member for local failure than is the average stress $S_{c(u)}$ (Ref. 55). A relation giving the ratio of maximum to average stress in the stiffener is

$$\frac{S_{c(u)(max)}}{S_{c(u)}} = 1 + 0.646\left(1.2 - \frac{b_1}{h_u}\right)(1 - n) \quad (4\text{-}48)$$

where

$S_{c(u)(max)}$ = maximum compressive stress in skin stiffener, lb/in.^{2}

h_u = free length of stiffener, in. (distance between centroids of flange-to-stiffener rivet pattern in riveted connections and distance between flange-to-stiffener welds in welded connections)

Eq. 4-48 is based on less supporting data than are other empirical relations of the Ref. 55 method and is, therefore, possibly not as reliable; but a more complete formulation of the ratio $S_{c(u)(max)}/S_{c(u)}$ is not available. Stiffeners working as columns gain lateral buckling resistance from the attached web and, therefore, require that an effective (reduced) column length L_e be used for strength calculations. This can be determined by means of the following empirical equation:

$$\left.\begin{aligned} L_e &= \frac{h_u}{\left[1 + n^2\left(3 - 2\dfrac{b_1}{h_u}\right)\right]^{1/2}} && (b_1 < 1.5H) \\ L_e &= h_u && (b_1 > 1.5H) \end{aligned}\right\}, \text{in.} \quad (4\text{-}49)$$

All terms are as defined for Eqs. 4-46 to 4-48. Column failure is attributed to symmetrical stiffeners, only, and is checked by comparing average compressive stress $S_{c(u)}$ to allowable

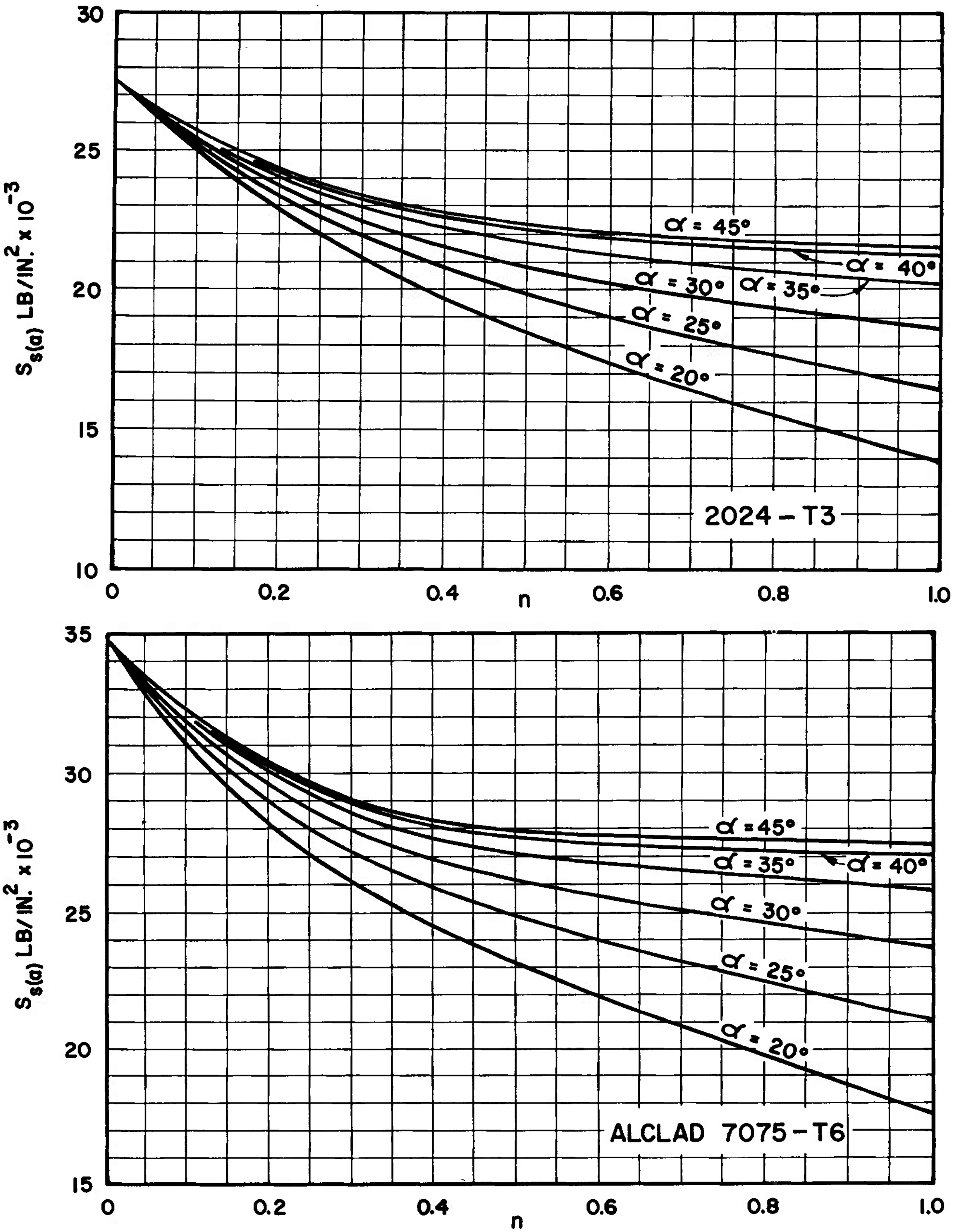


Figure 4-39. Allowable Web Shear Stress [55]

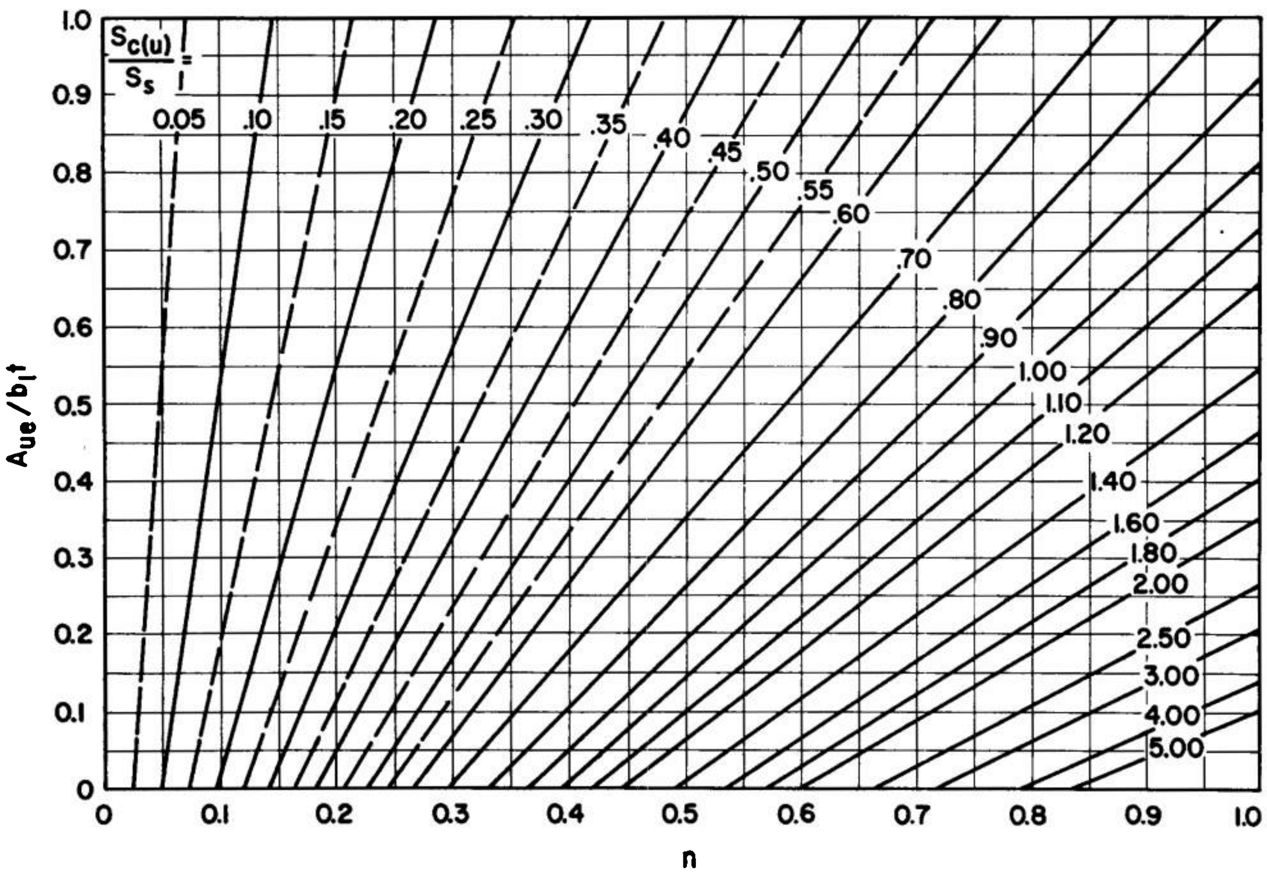


Figure 4-40. Diagonal Tension Analysis Chart[55]

compressive stress $S_{c(a)}$. Standard equations and curves of $S_{c(a)}$ for various materials (steel, aluminum, magnesium) and their alloys are available in many published engineering handbooks (see par. 4-1.2.2.3). Specific equations vary for different materials, alloys, shapes, and forming methods; although the total number of equations is very few. Since the Ref. 55 method is based on two specific aluminum alloys, only the equations for these materials and for specified forming methods are included here[5]. Similar equations for other materials or manufacturing means are given in the same source.

Short Column Formula

Use when the following relationships apply:

$$\left(\frac{L_e}{r}\right) > B_4\left(\frac{E}{S_y}\right)^{1/2} = \frac{B_5}{S_y^{1/2}}, \text{ dimensionless} \quad (4\text{-}50)$$

$$S_{c(a)} = S_y\left[1 - B_1\frac{\left(\frac{L_e}{r}\right)}{\pi\left(\frac{E}{S_y}\right)^{1/2}}\right] = S_y\left[1 - B_2\left(\frac{L_e}{r}\right)S_y^{1/2}\right], \text{ lb/in.}^2 \quad (4\text{-}51)$$

$$S_y = S_{cy}\,(1 + B_3 S_{cy}^{1/3}), \text{ lb/in.}^2 \quad (4\text{-}52)$$

Long Column Formula (Use when relationships of Eq. 4-50 do not apply.)

$$S_{c(a)} = \frac{\pi^2 E}{\left(\frac{L_e}{r}\right)^2} = \frac{B_6}{\left(\frac{L_e}{r}\right)^2}, \text{ lb/in.}^2 \quad (4\text{-}53)$$

where

$S_{c(a)}$ = allowable compressive stress, lb/in.^{2}

S_y = compressive vield stress for column failure, lb/in.^{2}

S_{cy} = compressive yield stress corresponding to 0.002 in. permanent strain (from standard specimen tests) S_{cy} = 42,000 lb/in.^{2} for 2024-T3 drawn shapes and 71,000 lb/in.^{2} for 7075-T6 extruded shapes, respectively (Ref. 6).

L_e/r = equivalent slenderness ratio of the column, dimensionless.

$B_1, B_2, B_3, B_4, B_5, B_6$ = constants to be used with Eqs. 4-50 to 4-53, Table 4-7.

Other terms are defined previously.

TABLE 4-7 CONSTANTS FOR EQUATIONS 4-50 TO 4-53

	2024-T2 *Alloy* *(Drawn Shapes)*	7075-T6 *Alloy* *(Extruded Shapes)*	*Units*
B_1	0.385	0.272	----
B_2	37.5×10^{-6}	26.7×10^{-6}	$(\text{lb/in.}^2)^{-1/2}$
B_3	10^{-3}	2×10^{-3}	$(\text{lb/in.}^2)^{-1/2}$
B_4	1.732	1.224	----
B_5	17.8×10^3	12.5×10^3	$(\text{lb/in.}^2)^{-1/4}$
B_6	105.6×10^6	103.6×10^6	$(\text{lb/in.}^2)^{-1/4}$

Stiffeners on One Side of Web Only

Stiffeners on only one side of the web are eccentrically loaded columns wherein eccentricity of load and, hence, critical stress depend on interaction of both web and stiffener. Theories for this complex problem are scarce, but Ref. 55 recommends that two checks be made on single stiffener design to prevent a columnlike (two half-wave) failure similar to true column failure of double stiffeners:

First Check:

Determine whether $S_{c(u)} < S_y$ (4-54)

Second Check:

Determine whether the average compressive stress across the stiffener cross section $S'_{c(u)}$ (from Eq. 4-55) is less than the allowable compressive stress $S'_{c(a)}$ determined by using $(h_u/2r)$ for the slenderness ratio (L_e/r) in Eqs. 4-51 or 4-53, whichever one applies; or expressed mathematically,

$$S'_{c(u)} = \frac{S_{c(u)}A_{ue}}{A_u} = \frac{S_{c(u)}}{1+\left(\frac{e}{r}\right)^2} < S'_{c(a)} \quad (4\text{-}55)$$

Forced-crippling failure of stiffeners, particularly if thin-sectioned, is an important consideration in stressed-skin structures. Crippling stress data for columns exist in material literature (e.g., Ref. 6), but these apply to simple columns, only—those uncomplicated by diagonal tension effects from the web. The Ref. 55 method gives empirical formulas for this situation; but they offer only a first approximation to the problem since the bottom of a ±20 percent scatter band of tests is represented. Other limitations of this method are (1) data given is based on stiffeners and skin of the same material, (2) only riveted attachment was used, (3) stiffeners had open cross sections, and (4) information is limited to the two aluminum alloys previously cited. Based on these limitations, the forced crippling equations can be conservatively extended to include closed section stiffeners and welded attachments.

$$S_o = B_7\, n^{2/3}\left(\frac{t_u}{t}\right)^{1/3},\ \text{lb/in.}^2 \quad (4\text{-}56)$$

where

S_o = forced crippling stress, lb/in.^{2}

B_7 = constant depending on type of material and stiffener, dimensionless

$B_7 = 21\times10^3$ and 26×10^3 for double and single stiffeners, respectively (2024-T3 alloy)

$B_7 = 26\times10^3$ and 32.5×10^3 for double and single stiffeners, respectively (7075-T3 alloy)

If S_o is greater than the proportional limit of the material, a plasticity correction (similar to Fig. 4-34) is necessary by means of a compression stress-strain diagram. The criterion for preventing failure by forced crippling is

$$S_{c(u)(max)} < S_o \quad (4\text{-}57)$$

Strength of Flanges

Primary bending stress

$$S_{b1} = \pm\frac{My}{I_F},\ \text{lb/in.}^2 \quad (4\text{-}58)$$

where

M = primary bending moment existing on flange section, in.-lb

y = distance from neutral axis of beam to a given fiber of flange, in.

I_F = moment of inertia of flanges about beam neutral axis (web not included), in.^{4}

± sign corresponds to tension and compression flanges, respectively (ie., S_{TB1} and S_{CB1})

Average compressive axial stress due to horizontal component of diagonal tension (on compression and tension flanges, respectively)

$$\left.\begin{aligned} S_{CC} &= -\frac{nV\cot a}{2A_{CF}}, \text{ lb/in.}^2 \\ S_{CT} &= -\frac{nV\cot a}{2A_{TF}}, \text{ lb/in.}^2 \end{aligned}\right\} \quad (4\text{-}59)$$

where

a = angle between direction of diagonal tension and neutral axis of beam, deg

A_{CF}, A_{TF} = areas of compression and tension flange cross sections, respectively, in.^{2}

Secondary bending stress due to vertical components of diagonal tension (on compression and tension flanges, respectively)

$$\left.\begin{aligned} S_{CB2} &= \pm\frac{M'y'}{I_{CF}} = \pm\frac{C_3 nS_s t b_1^2 y'}{12I_{CF}}, \text{ lb/in.}^2 \\ S_{TB2} &= \pm\frac{C_3 nS_s t b_1^2 y'}{12I_{TF}}, \text{ lb/in.}^2 \end{aligned}\right\} \quad (4\text{-}60)$$

where

M' = secondary bending moment existing on flange section (max at stiffeners), in.-lb

y' = distance from neutral axis of flange section to a given flange fiber, in.

I_{CF}, I_{TF} = moments of inertia of compression and tension flanges, respectively, about their own neutral axes, in.^{4}

C_3 = stress concentration factor, dimensionless (analogous to C_2 for the skin described earlier in the subparagraph) (see Fig. 4-37).

± sign corresponds to fibers on the outboard and inboard sides of the flange neutral axis with respect to beam neutral axis

Maximum stress condition on compression and tension flanges, respectively, limited to less than material yield stress is

$$\left.\begin{aligned} S_{c(max)} &= (S_{CB1} - S_{CB2})_{max} + S_{CC}, \text{ lb/in.}^2 \\ S_{T(max)} &= S_{TB1} + S_{TB2} + S_{CT}, \text{ lb/in.}^2 \end{aligned}\right\} \quad (4\text{-}61)$$

Note that correct summing of S_{CB1} and S_{CB2}, and S_{TB1} and S_{TB2}, is possible only for the same flange fiber (see an example in par. 4-2.3.3.3). Much additional information is contained in Ref. 55—especially on strength of rivets—and should be consulted for details. The Ref. 55 method as presented here can be usefully extended to automotive applications but, at present, it possesses certain shortcomings. The principal of these relate to the limited number of materials, beam loadings, and jointing methods covered in the report. Automotive-oriented research into these and other areas of the incomplete diagonal tension theory will undoubtedly extend its usefulness.

4-2.3.3.3 Design Calculations

Consider the panel shown in Fig. 4-30(B) to be a wall panel of an automotive vehicle, and that it is subjected to the loads P_1 and P_2, each equal to 20,000 lb, acting at the points indicated. The reactions R_1 and R_2 are 21 ft apart. Thus, the maximum applied vertical shear V occurs between R_1 and P_1 and between R_2 and P_2; and the maximum applied bending moment occurs between P_1 and P_2. A simple block diagram of the panel, together with sections through the stiffener and flanges, is shown in Fig. 4-41. Other information pertinent to the panel, and to the calculations, are as follows (see par. 4-2.3.3.2 and Fig. 4-41 for explanation of symbols):

Material: 2024-T3 aluminum skin and drawn rectangular cross section tube for stiffeners and flanges

$$E = 10.5 \times 10^6 \text{ lb/in.}^2$$

$S_{ty} = S_{cy} = 42{,}000$ lb/in.^{2}

(yield stress in tension and compression, respectively)

Compression flange (upper): 3 in. X 3 in. tube

Tension flange (lower): 3 in. X 5 in. tube

Stiffeners: 1 in. X 2 in. tube on one side of skin-web only

h = 63.5 in.

H = 68 in

b = 20 in.

b_1 = 21 in.

t_u = 0.0625 in.

t = 0.020 in.

h_u = 63.5 in.

t_F = 0.075 in.

Preliminary Calculations

a. Properties of Stiffeners

$$I_u = \frac{(1.0)(2)^3 - (0.875)(1.875)^3}{12} = 0.187 \text{ in.}^4$$

$$A_u = (1.0)(2) - (0.875)(1.875) = 0.36 \text{ in.}^2$$

$$e = 1.00 + 0.01 = 1.01 \text{ in.}$$

$$r = \left(\frac{0.187}{0.36}\right)^{1/2} = 0.72 \text{ in.}$$

$$A_{ue} = \frac{A_u}{1 + \left(\frac{e}{r}\right)^2} = \frac{0.36}{1 + \left(\frac{1.01}{0.72}\right)^2} = 0.121 \text{ in.}^2$$

[see Eq. 4-47]

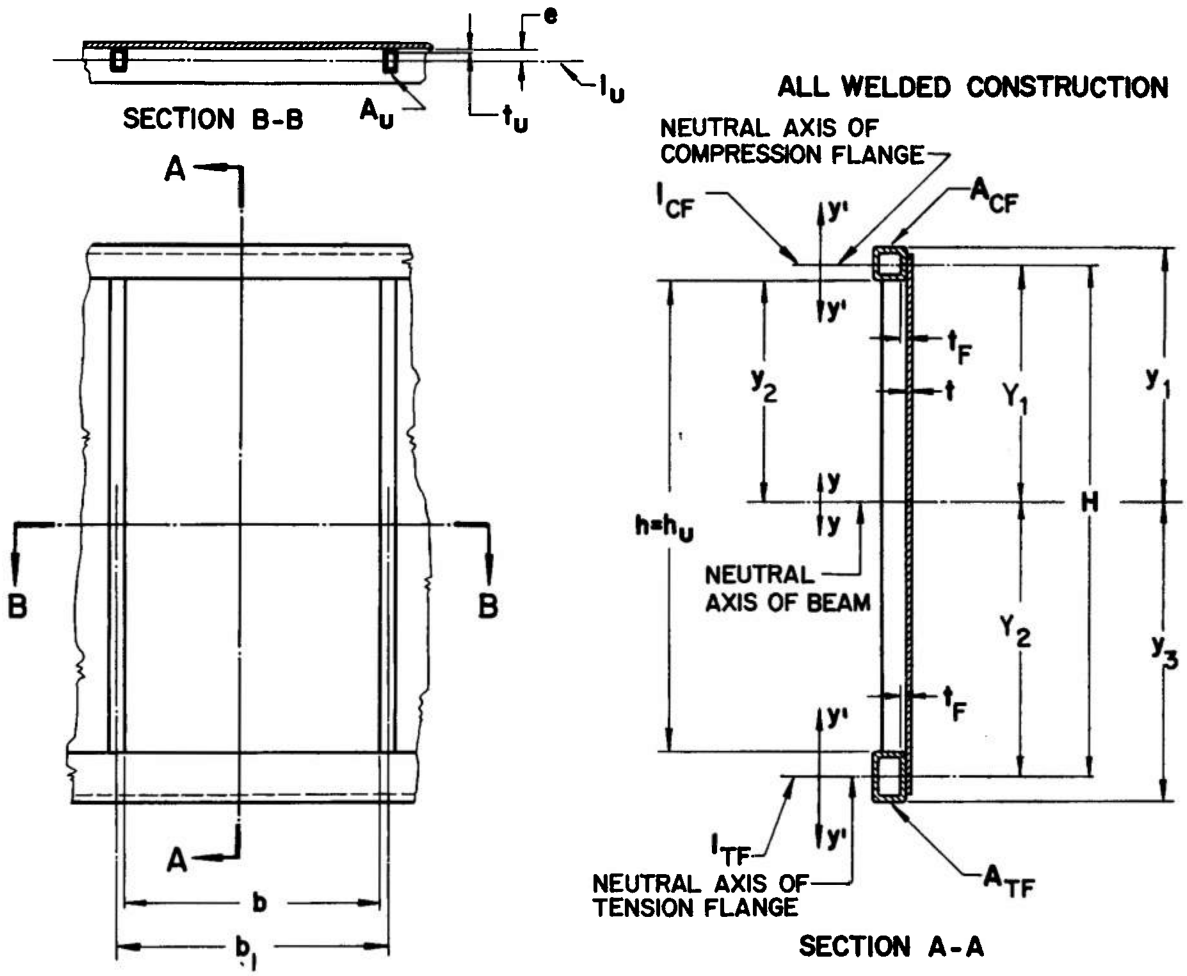


Figure 4-41. Stressed-skin Structure, Nomenclature for Example Problem

b. Properties of Flanges

$$I_{TF} = \frac{(3)(5)^3 - (2.85)(4.85)^3}{12} = 4.16 \text{ in.}^4$$

$$I_{CF} = \frac{(3)^4 - (2.85)^4}{12} = 1.25 \text{ in.}^4$$

$$A_{TF} = (3)(5) - (2.85)(4.85) = 1.2 \text{ in.}^2$$

$$A_{CF} = (3)^2 - (2.85)^2 = 0.88 \text{ in.}^2$$

c. Properties of the Beam (Skin not included)

$$\left.\begin{aligned} Y_1 &= H\left(\frac{A_{TF}}{A_{TF} + A_{CF}}\right) = H - Y_2 \\ &= 39.2 \text{ in.} \\ Y_2 &= H\left(\frac{A_{CF}}{A_{TF} + A_{CF}}\right) \\ &= 68\left(\frac{0.88}{1.2 + 0.88}\right) = 28.8 \text{ in.} \end{aligned}\right\} \text{[see Fig. 4-41]}$$

$$\begin{aligned} I_F &= 4.16 + 1.2(39.2)^2 + 1.25 \\ &\quad + 0.88(28.8)^2 \\ &= 2580 \text{ in.}^2 \end{aligned} \quad \text{(parallel axis theorem)}$$

Elastic Shear Buckling Stress of Skin

$$\frac{h}{b} = \frac{63.5}{20} = 3.18, \quad K_{ss} = 5.35 \qquad \text{[Fig. 4-33(A)]}$$

$$\left.\begin{aligned} \frac{t_u}{t} &= \frac{0.0625}{0.020} = 3.13 \\ \frac{t_F}{t} &= \frac{0.075}{0.020} = 3.75 \end{aligned}\right\} R_h = R_b = 1.62 \text{ [maximum values, Fig. 4-33(B)]}$$

$$\begin{aligned} S_{s(cre)} &= 5.35(10.5 \times 10^6)\left(\frac{0.020}{20}\right)^2 (1.62 + 0) \\ &= 91 \text{ lb/in.}^2 \end{aligned} \qquad \text{[Eq. 4-41]}$$

$$S_{s(cre)} = S_{s(cr)} \qquad \text{[Fig. 4-34]}$$

Total Skin Shear Stress

$$S_s = \frac{20{,}000}{68(0.020)} = 14{,}700 \text{ lb/in.}^2 \qquad \text{[Eq. 4-43]}$$

Loading Ratio

$$\frac{S_s}{S_{s(cr)}} = \frac{14{,}700}{91} = 162$$

then, $n = 0.80$ [Fig. 4-35]

$S_{c(u)}/S_s$ Ratio and Diagonal Tension Angle

$$\frac{A_{ue}}{b_1 t} = \frac{0.121}{21(0.020)} = 0.288, \text{ dimensionless}$$

(see preliminary calculations)

$$\frac{S_{c(u)}}{S_s} \cong 1.52 \qquad \text{[Fig. 4-40]}$$

$$\tan \alpha = 0.74 \qquad \text{[Fig. 4-38]}$$

Maximum Skin Shear Stress

$C_1 = 0.044$, dimensionless [Fig. 4-36]

$$d = 0.7(21)\left[\frac{0.020}{68(4.16 + 1.25)}\right]^{1/4} \qquad \text{[Eq. 4-45]}$$

$= 1.26$, dimensionless

then $C_2 \approx 0.02$ [Fig. 4-37]

$$\begin{aligned} S_{s(max)} &= 14{,}700\left[1 + (0.80)^2(0.044)\right]\left[1 + 0.80(0.02)\right] \\ &= 15{,}300 \text{ lb/in.}^2 \end{aligned} \qquad \text{[Eq. 4-46]}$$

Allowable Skin Shear Stress

$$S_{s(a)} = 21{,}900 \text{ lb/in.}^2 \qquad \text{[Fig. 4-39]}$$

$$MS = \frac{21.9}{15.3} - 1 = 0.43$$

No reduction in $S_{s(a)}$ is made, however, due to the concentrated load P on the skin (see theory). A thinner skin would result in a higher working stress but would also produce more noticeable ripples. In addition, the greater thickness will enable the skin to better resist accidental punctures. The 0.020-in. thickness is, therefore, retained.

Average Compressive Stress in Skin Stiffener

$$S_{c(u)} = 1.52 S_s = 22{,}350 \text{ lb/in.}^2 \qquad \text{[Fig. 4-40]}$$

or

$$S_{c(u)} = \frac{0.80\ (14{,}700)\ (0.74)}{0.288 + 0.5\ (1 - 0.80)} = 22{,}400\ \text{lb/in.}^2$$ [Eq. 4-47]

Maximum Compressive Stress in Skin Stiffener

$$\frac{S_{c(u)(max)}}{S_{c(u)}} = 1 + 0.646\left(1.2 - \frac{21}{63.5}\right)(1 - 0.80)$$

$$= 1.11$$ [Eq. 4-48]

then

$$S_{c(u)(max)} = 1.11\ (22{,}400) = 25{,}000\ \text{lb/in.}^2$$

Compressive Yield Stress for Primary Failure of Stiffener

$$S_y = 42{,}000\left[1 + 10^{-3}(42{,}000)^{1/2}\right]$$ [Eq. 4-51]

$$= (42{,}000)1.205$$

$$S_y = 50{,}600\ \text{lb/in.}^2$$

Stiffener Strength for Primary Failure

$$\frac{S_{c(u)}}{S_y} = \frac{22{,}400}{50{,}600} = 0.443$$

Therefore;

$S_{c(u)} < S_y$, and "First Check" is satisfied.

$$\frac{b_1}{1.5H} = \frac{21}{102}$$ [Eq. 4-54]

Therefore; $b_1 < 1.5H$, and

$$L_e = \frac{63.5}{\left[1 + 0.80^2\left(3 - \frac{2 \times 21}{63.5}\right)\right]^{1/2}}$$ [Eq. 4-49]

$$L_e = 28.9\ \text{in.}$$

$$\frac{L_e}{r} = \frac{28.9}{0.72} = 40.15$$

$$\frac{B_5}{S_y^{1/2}} = \frac{17.8 \times 10^3}{(5.06 \times 10^4)^{1/2}} = 79.1$$ [Eq. 4-50]

$$\frac{\frac{L_e}{r}}{\frac{B_5}{S_y^{1/2}}} = \frac{40.15}{79.1}$$

Therefore,

$\frac{L_e}{r} < \frac{B_5}{S_y^{1/2}}$, and the Short Column Formula applies. [Eq. 4-51]

Since stiffeners are placed on one side of the skin-web only, $S'_{c(a)}$ instead of $S_{c(a)}$ is calculated from modified Eq. 4-51 (see discussion on "Second Check").

$$\frac{h_u}{2r} = \frac{63.5}{2\ (0.72)} = 4.41$$

$$S'_{c(a)} = 50{,}600\ (1 - 37.50 \times 10^{-6} \times 44.1 \times 50{,}600^{1/2})$$

$$= 50{,}600 \times 0.628$$ [Eq. 4-51]

$$= 31{,}800\ \text{lb/in.}^2$$

$$S'_{c(u)} = \frac{22{,}400}{1 + \left(\frac{1.01}{0.72}\right)^2} = 7{,}540\ \text{lb/in.}^2$$ [Eq. 4-55]

$$\frac{S'_{c(u)}}{S_{c(a)}} = \frac{7{,}540}{31{,}800} = 0.237$$

Therefore, $S'_{c(u)} \ll S'_{c(a)}$, and "Second Check" is satisfied. [Eq. 4-55]

Thus, it is evident that the stiffener is very safe from primary column failure according to the two recommended checks.

Stiffener Strength for Secondary Failure

$$S_o = 26 \times 13^3 (0.80)^{2/3} (3.13)^{1/3}$$ [Eq. 4-56]

$$= 32{,}700\ \text{lb/in.}^2$$

S_o is slightly above the compressive proportional limit, but plasticity correction is not significant.

$$\frac{S_{c(u)(max)}}{S_o} = \frac{25{,}000}{32{,}700} = 0.765$$

Therefore $S_{c(u)(max)} < S_o$ [Eq. 4-57]

$$MS = \frac{32.7}{25.0} - 1 = 0.30$$

Flange, Primary Bending Stress

$$M_{(max)} = P\frac{\ell}{3} = 20{,}000\left(\frac{21}{3}\right) \quad (\ell = \text{distance between reactions, ft})$$

$$= 140{,}000 \text{ ft-lb}$$

$$\left.\begin{aligned} y_1 &= Y_1 + 1.5 = 28.8 + 1.5 = 30.3 \text{ in.} \\ y_2 &= Y_1 - 1.5 = 27.3 \text{ in.} \\ y_3 &= Y_2 + 2.5 = 39.2 + 2.5 = 41.7 \text{ in.} \end{aligned}\right\}$$

[Fig. 4-41]

$$S_{TB1} = \frac{1.4\times10^5(12)(41.7)}{2.58\times10^3} = 27{,}200 \text{ lb/in.}^2$$

(tension flange, outermost fiber)

$$S_{CB1} = \frac{1.4\times10^5(12)(30.3)}{2.58\times10^3} = -19{,}700 \text{ lb/in.}^2$$

(compression flange, outermost fiber)

$$S'_{CB1} = -19{,}700\left(\frac{27.3}{30.3}\right) = -17{,}750 \text{ lb/in.}^2$$

(compression flange, innermost fiber)

[Eq. 4-58]

Flange, Average Compressive Axial Stress

$$\left.\begin{aligned} S_{CT} &= -\frac{0.80\,(20{,}000)\,(1.35)}{2(1.2)} = -9{,}000 \text{ lb/in.}^2 \\ S_{CC} &= -\frac{0.80\,(20{,}000)\,(1.35)}{2(0.88)} = -12{,}300 \text{ lb/in.}^2 \end{aligned}\right\}$$

[Eq. 4-59]

Flange, Secondary Bending Stress

$C_3 = 0.99$ (since $d = 1.26$ by previous calculation) [Fig. 4-37]

$$S_{TB2} = \pm\frac{0.99\,(0.80)(14{,}700)(0.020)(21)^2(2.5)}{12(4.16)} \quad \text{[Eq. 4-60]}$$

$$= \pm5{,}140 \text{ lb/in.}^2$$

$$S_{CB2} = \pm\frac{0.99(0.80)(14{,}700)(0.020)(21)^2(1.5)}{12(1.25)} \quad \text{[Eq. 4-60]}$$

$$= \pm5{,}140\left(\frac{1.5}{2.5}\right)\left(\frac{4.16}{1.25}\right)$$

$$= \pm10{,}280 \text{ lb/in.}^2$$

Maximum Stress on Tension Flange

$$S_{T(max)} = 27{,}200 + 5{,}140 - 9{,}000$$

$$= 23{,}340 \text{ lb/in.}^2 \text{ (outermost fiber)}$$

Maximum Stress on Compression Flange

$$S_{c(max)} = (-17{,}750 - 10{,}280) - 12{,}300$$

$$= -40{,}300 \text{ lb/in.}^2 \text{ (innermost fiber)}$$

[Eq. 4-61]

Note, investigation of the outermost fiber, only, gives a misleading maximum stress of $-21{,}700$ lb/in.^{2} for the compression flange because of the tension component $+S_{CB2}$.

Margins of Safety (based on yield stress)

$$MS_T = \frac{S_{ty}}{S_{T(max)}} - 1 = \frac{42{,}000}{23{,}340} - 1 = 0.80 \text{ (tension flange)}$$

$$MS_C = \frac{S_{cy}}{S_{c(max)}} - 1 = \frac{42{,}000}{40{,}300} - 1 = 0.04 \text{ (compression flange)}$$

The foregoing example is not intended to show the best possible design, but only to illustrate procedures. One obvious improvement, for example, would be to better equalize flange stresses by making the compression flange deeper and the tension flange shallower.

4-3 LOAD DISTRIBUTION

4-3.1 BALANCE CONSIDERATIONS

Load distribution with regard to balance is an important consideration for proper vehicle performance. Distribution techniques are closely related to design weight and center-of-gravity location on a particular vehicle. In short, design for balance must ensure that vehicle component and cargo weights are distributed so as to result in a stable and favorable center-of-gravity

position for expected types of operation. It should not be implied that stability is the sole consideration of load distribution for balance. Castastrophic tipping need not occur before detrimental effects take place. Serious loss of performance can result without stability problems if planned and actual operating centers of gravity differ significantly. Empty weight, gross vehicle weight, and their respective centers of gravity are specified or carefully set down during preliminary design. However, they become cumbersome to maintain or even to keep track of without a weight-balance recording system as design moves into the final and detail stages. Weight and load control criteria are prone to many obstacles without a fully established and continuously maintained system of data reporting. Failure to realize its importance, lack of communication between supporting departments, and insufficient analysis for need or function of structures and equipment are some reasons to frustrate a weight and balance accounting system that is not officially part of design policy. Of course, any program of this type will result in extra design and production costs, yet it is more and more necessary with increasingly sophisticated military vehicles.

Ref. 61 discusses the importance of, and some of the requirements for, balance in automotive design. Although no less important in the automotive field, weight-balance accounting methods are much more firmly established in aircraft design mainly because of greater, inherent necessity. Aircraft-oriented weight-balance systems are sources from which similar techniques can be evolved for land vehicles. MIL-STD-254[62] gives a detailed data reporting system for aircraft which, while not directly applicable to automotive use, is still of interest to designers in the automotive field. Basically, any weight-balance system must break down structures and equipment into logical groups and subgroups, note weights of all meaningful components within groups, and account for their distribution via increments or stations relative to the vehicle. Considerable flexibility needs to be built into a system to allow its use for different vehicle types.

Balance problems of specific concern to the body designer are due to cargo- and equipment-imposed loads. To accommodate these primary load sources, assumption of expected or typical magnitudes has to take place during the design procedure. Equipment loads allow the designer to deal in a rather well-defined region. Numbers, types, sizes, weights, and locations of permanently or semipermanently placed equipment are known or calculable entities. Cargo loads, on the other hand, vary from a somewhat well-defined environment on combat vehicles (i.e., crew, munitions) to a much less-defined situation on, say, general cargo vehicles. Here, the designer must work in an area where the variety of cargo is great, overload is quite possible, and even the position of "safe" payload within the body or hull becomes significant. Yet with such unknowns, the design must be based on typical loads. To make this a livable arrangement, real effort should be made to collect and compile loading data from which additional design criteria can be formulated relative to vehicle weight balance. If the number of unknowns concerning loads is reduced, a sizable design gap will have been narrowed.

4-3.2 DISTRIBUTION TO REDUCE LOAD CONCENTRATION

Load concentration occurs in vehicle structures whenever substantial forces are allowed to react against body or hull members through small contact areas. Actually, it is the magnitude of the quantity $S = P/A$ (the very basic normal stress relation of mechanics) that determines how damaging the effect of a given load is. Very small forces acting over sufficiently small surface areas can be shown to be destructive by this relation. Such problems are interesting and have application in certain fields; but in automotive structures design, it is the substantial forces and the sizable bearing areas necessary to contain them that are of importance.

Suspension loads and cargo loads are the main contributors to body and hull load concentration. These concentrated load sources exist in both static (vehicle at rest) and dynamic forms (vehicle in motion). Obviously, the dynamic loads will be more severe and, therefore, are of great interest in vehicle design. These same loads are the most difficult to calculate, however, dynamic suspension loads

originate from reactions between terrain and ground-contacting elements of the vehicle. It is the many unknowns of terrain-imposed loads, particularly in analysis, that make this area one of the most complex in vehicle design. Calculation of dynamic cargo loads has some similar difficulties but here analysis is better established. Dynamic loads, in particular, are discussed in par. 4-1.4. Treatment of load concentrations will, therefore, be based on static loads even though they are of less concern to the design engineer. This is done because of available information, ease of illustrating principles, and because the same load-distribution methods are used.

Suspension loads enter an automotive structure through suspension-mounting points or through body-to-frame mounting points, depending on the chassis-to-body (or hull) relationship of a particular vehicle (par. 1-4). Either system, discussed in the paragraphs which follow, will result in a very limited number of points for reactions to be absorbed in the body.

a. *Suspension Mounting Points.* The number of suspension mounts is generally proportional to the number of road-contacting elements—i.e., wheels, road wheels, bogies, and sprockets—but will vary with suspension system and vehicle type. One- and two-axle wheeled vehicles suffer great disadvantage from the standpoint of minimum available load absorption points. Three- and four-axle, independently suspended, wheeled vehicles derive beneficial results from having an increased number of suspension mountings. This concept is continued and is used to considerable advantage in track-laying vehicles where the number of road-contacting components can be substantially increased, and on some designs is limited only by the space available. A similar increase of wheels on wheeled vehicles becomes prohibitive due to an introduction of many additional design and maintenance problems as well as to a substantial cost increase. However, it is not inconceivable for future wheeled military vehicles to have a similar number of ground-contacting elements as found on tracked vehicles now in use.

b. *Body-to-frame Mounting Points.* The number of body-to-frame mounts is also limited to allow for independent frame deflection and to isolate the body from many severe loads and noises. Thus, the primary cause of concentrated suspension loads is the limited number of reaction points in automotive structures.

Tracked vehicles absorb suspension loads by means of suspension mounts or support housings mounted directly to their hulls or to auxiliary suspension mounting structures (par. 3-27). The use of one method or the other depends on a specific hull construction. Support housings can be bolted to hulls as in Fig. 2-7 or welded to them as shown by Fig. 2-8. The mountings actually make contact with the hull at massive pads provided for this purpose and made with as large surface areas as practicable to distribute load.

Wheeled vehicles employ constructions quite similar to those in tracked vehicles to contain suspension loads. In short, specially selected structural members—frames, frame-like components, supports, and stiffeners—are used as the concentrated load carriers by preferential location of these members. Bushings, seats, and padded surfaces of rubber and other elastomeric materials also find application in this design area. Fig. 1-16(B) shows clear examples of suspension load distribution methods in the body of a ¼-ton utility truck. The main channel rails of the underbody pass over the rear axle and suspension mounting points; other structural shapes pass over and form the rear spring seats.

Cargo loads can act in a concentrated manner on vehicle floors, although many cargo types produce inherently distributed loading effects. Pars. 4-5.1 and 4-7.1 consider the types of cargo loads in detail. Equipment-type cargo loads are not considered here. Many such items are permanently or semipermanently placed in vehicles, and load concentrations due to them can be alleviated in a manner previously described for suspension loads in wheeled vehicles. Transported cargo, however, can and will be placed anywhere on a cargo floor. Any loadable portion of the floor must have sufficient strength for localized high loads. Strength may come from either the floor itself, from secondary cushioning and protective flooring (pars. 4-4.1 and 4-4.2), or from similar cushioning pads placed under cargo packages and containers. It is interesting to note in connection with cargo packaging and containerization that such provisions may quite often cause load concentration to occur with cargos that naturally exhibit uniform load distribution. A

standard liqu'd drum with its narrow rim floor contacting surface and a load of bricks, palletized to facilitate handling but thus responsible for greatly reduced floor contact area, are offered as examples. To illustrate the difference between natural and concentrated load intensities, consider the following two example problems.

Example 1

Cargo: petroleum in 55-gal drum containers
Typical drum dimensions:

ID : d_i = 22½ in.
Rim mean dia : d_m = 22¾ in.
Approximate rim width : t = 3/16 in.
Density of oil : $\rho = 0.82(62.4) = 51.2$ lb/ft^3 $= 0.0296$ lb/in.^{3}
Volume of oil per drum: $V = 55(231) = 12{,}700$ in.^{3}
Weight of oil per drum: $P = V\rho = 12{,}700 = 12{,}700(0.0296) = 376$ lb

Natural contact area of load (without drum rim):

$$A = \pi\left(\frac{d_i}{2}\right)^2 = \pi\,(11.25)^2 = 398 \text{ in.}^2$$

Actual contact area of load (rim area);

$$A' = \pi d_m\, t = \pi(22.75)(0.1875) = 13.4 \text{ in.}^2$$

Natural load intensity S: $\dfrac{P}{A} = \dfrac{376}{398} = 0.945$ lb/in.^{2}

Actual load intensity $S' = \dfrac{P}{A'} = \dfrac{376}{13.4} = 28$ lb/in.^{2}

Load intensity ratio $R = \dfrac{S'}{S} = \dfrac{28}{0.945} = 29.6$

The raised drum floor will deflect when the container is filled with fluid. Such an effect was not considered in the example, but this would contribute to the contact area and would reduce the actual load intensity S . Nevertheless, this simplified analysis shows significant load concentration. Compare the magnitude of load intensity of that in the example problem of par. 4-2.3.2.

Example 2

Cargo: pallet loads of brick stacked into units with dimensions 40 in. by 60 in. by 45 in. high. The pallet base is composed of three two-by-fours (with the larger dimension perpendicular to the floor) placed symmetrically under the load.
Then:

$$\rho = 125 \text{ lb/ft}^3 = 0.724 \text{ lb/in.}^3$$

$$V = (40)(60)(45) = 108{,}000 \text{ in.}^3$$

$$P = V\rho = 108{,}000(0.0724) = 7820 \text{ lb}$$

$$A = 40(60) = 2400 \text{ in.}^2\ ;\ A' = 3(60)(2) = 360 \text{ in.}^2$$

$$S = \frac{7820}{2400} = 3.26 \text{ lb/in.}^2\ ;\ S' = \frac{7820}{360} = 21.7 \text{ lb/in.}^2$$

$$R = \frac{21.7}{3.26} = 6.66$$

Substantial load concentration is shown in this example, also. Note that a still more severe condition can be caused by placing the pallet footing parallel to the shorter base dimension of the load. See the example in par. 4-2.3.2 for a comparative load intensity.

The significant point in the methods to reduce load concentration is to present as much contact area as practicable against the input forces—making the ratio of load to contact area, or elementary stress, small. How small is dictated by the allowable or design stress for the material. Practical application of even as basic a physical relation as $S = P/A$ is not easily accomplished, however, the assumption of uniform force distribution over the *entire* contact area is definitely a limiting factor. In practice, many nonideal conditions exist at the floor-to-cargo interface; relative stiffnesses, roughnesses, and material types of contacting surfaces are just some influencing factors. Therefore, proper load distribution methods for cargo involve much more than providing sufficient bearing area. An optimum distribution system would provide a very stiff pad of required area directly under the load with a substantially "softer" (elastomeric) layer of material under this as the primary floor contacting element. Thickness and deflection rate of the elastomer would be a necessary minimum of points for design analysis. These techniques are widely used for vibration and shock isolation in many other body and hull applications (pars. 3-2 and 3-28). Proper use in

the cargo area can cut down load intensities to their natural levels (example problems 1 and 2) or even below this—as determined by the allowable size of the distributing pad.

4-3.3 LOAD DISTRIBUTION TO REDUCE STRESS CONCENTRATION

Stress concentration in load-bearing members is said to occur when an abrupt change in stress level is noted between two points separated by only a small distance. Locations of such irregular stress patterns and relative maximum stress points coincide with positions of geometric irregularities found in a member. The distinguishing feature of true stress concentration is that maximum or actual stress will be greater than nominal stress as calculated by standard formulas—even though the net critical section of the member is used in the calculations. The ratio of actual stress to nominal stress is called the stress concentration factor. Furthermore, heavily concentrated loading is not a requirement to failure if the magnitude of stress concentration is sufficiently high. Thus, substantially different considerations apply here than for the load concentrations discussed in par. 4-3.2, which are also accompanied by stress concentrations of a sort, but not true stress concentrations as the term is used here.

Classical examples of stress concentrators, or stress raisers as they are commonly known, are notches in shafts and bars, holes in sheets and plates, screw threads, and even toolmarks—intended or accidental. Although sources of stress concentration are plentiful in automotive bodies and hulls, they are often not as readily evident on such interconnected structures as they are on individual parts. Examples of stress concentrations in the vehicle body occur in bolted, riveted, and welded joints; in sheet metal parts formed by bending, stamping, and contouring (due to thinning out or wrinkling of sections in production); and in flat sheets and plates containing lightening holes, cutouts, windows, doors, hatches, and other openings.

It is not strictly correct to include discussion of stress concentration reduction methods under the broader heading of load distribution, because the former is almost entirely a function of structural shape or form. General loading recommendations for areas of stress concentration in bodies or hulls would be to reduce or eliminate loads; however, such design freedom rarely exists and, in any case, would be skirting the issue—since stress concentration is independent of load distribution. A more reasonable approach is to investigate the nature of stress concentrations and reduce them through improved designs that reflect better loading capability.

Stress concentration under static loading has little importance in most cases, but there are exceptions to this rule of thumb[63]. Brittle materials have to be safeguarded from abrupt section variations even for static load applications. Fluctuating stresses require serious examination of shape for stress raisers in most materials[64]. The design engineer may continue his stress analysis using standard equations and then make use of well-established stress concentration factors to arrive at final results. For isotropic materials, these factors are functions of geometry only, but type of loading will influence magnitudes of factors. Stress concentration factors have been determined for a large number of cases that are of practical use in design; analytical as well as experimental methods (photoelasticity, strain gage measurements, etc.) have been used to obtain these values. The effects of notches, grooves, fillets, slots, and holes, occurring singly or in various arrays, have been determined for a variety of members and loadings. Stress concentration factors are available in numerous publications (e.g., Refs. 63—66). Ref. 66 is a noteworthy and extensive source devoted entirely to such factors.

Stress concentrations in automotive structures can be greatly reduced by adequate attention to design details. In items fabricated by bending, the following recommendations are important:

a. *Bend radii.* Observe minimum allowable bend radii. Necked down sections and skin fractures resulting from sharp bend radii are stress raisers (Fig. 4-42). Ref. 67 gives bend allowance tables for a number of metal and alloy sheets. A simple method for determining location of bends and length of stock required for bending sheet steel appears in Fig. 4-43.

b. *Fillets.* In machined and especially in cast parts, fillet radii should be as large as practicable. Castings develop stress concentrations at sharp corners and at abrupt

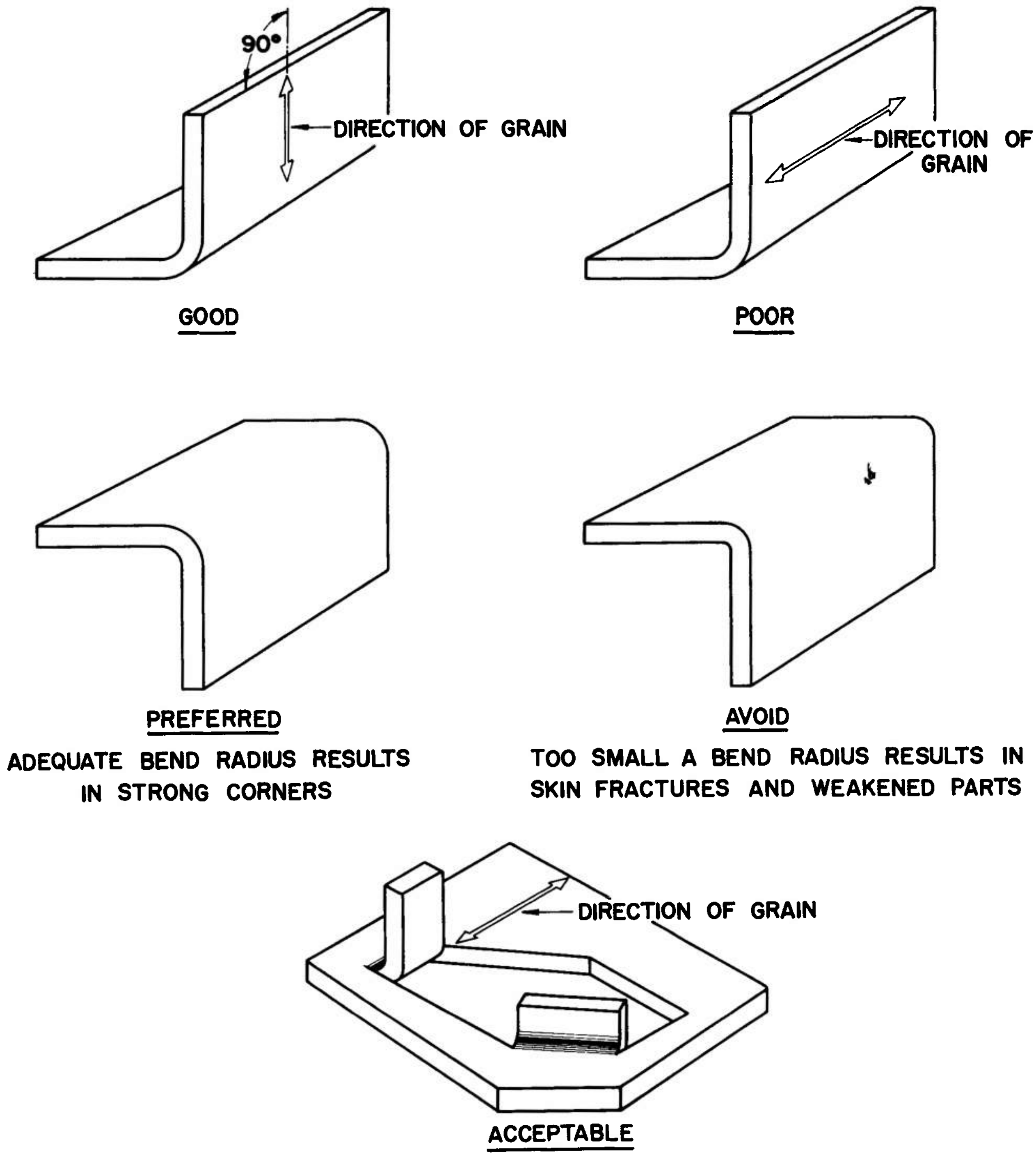
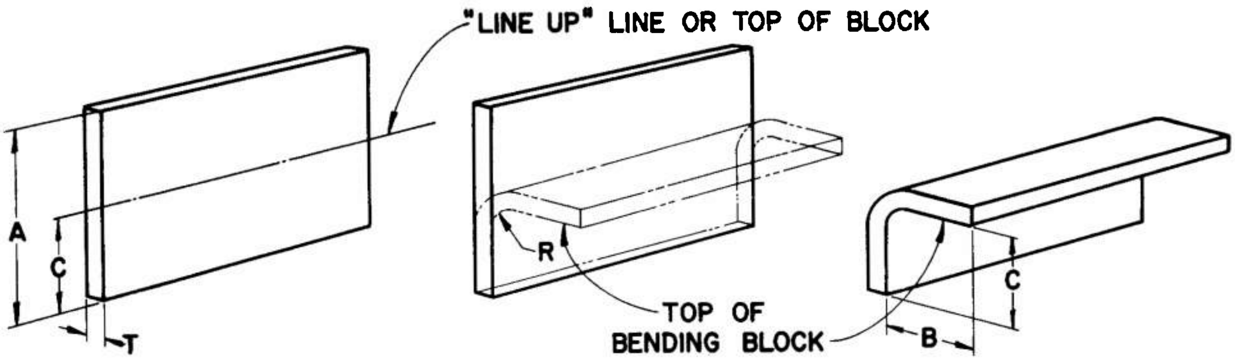


Figure 4-42. Radii and Grain Direction Considerations in Bending Sheet Metal



A = LENGTH OF STOCK REQUIRED
A = B + C + T/2 WHEN R = T (STEEL)
A = B + C WHEN R = 2 T (2017)
A = B + C WHEN R = 3 T (2024)
T = THICKNESS OF MATERIAL

Figure 4-43. Bends in Sheet Metal

changes in section due to metal shrinkage during their transition from the molten to the solid state. One method of reducing stress concentrations in castings through the use of generous radii fillets is shown in Fig. 4-44.

c. *Grain direction.* Orient bending in the same direction as grain of the material. Fig. 4-43 illustrates additional consideration for bend direction.

d. *Welds.* Welded assemblies are subject to stress concentrations due to abrupt changes in cross section at joints[68]. Such concentration can be substantially reduced in some cases by assuring proper fusion between parts to be joined and the weld metal. Improper fusion causes sharp notches to form at point *A* of the butt weld (Fig. 4-45(A)). However, even with good quality welds, stress raisers occur where the path of the force across a joint becomes substantially distorted. Locations of some concentrations as a result of this path distortion are at the reinforcement in the butt joint (point *B*, Fig. 4-45(A)) and at the toe and heel of the fillet joint (points *A* and *B*, Fig. 4-45 (B)). Expected values of the stress concentration factor for typical welds are given in Table 4-8.

Similar considerations apply to bolted or riveted joints and to holes and cutouts in plates or sheets. For these situations, as for any case of stress concentration, the basic cause should be

TABLE 4-8 STRESS CONCENTRATION FACTOR k FOR WELDS[68-70]

Location	k
Reinforced butt weld	1.2
Toe of transverse fillet weld	1.5
End of parallel fillet weld	2.7
T-butt joint with sharp corners	2.0

remedied. Sudden and irregular cross section changes in members sometimes cannot be completely eliminated, but very often they can be significiantly improved.

4-3.4 LOAD DISTRIBUTION TO REDUCE DEFLECTION

The rigidity of any automotive structure is relative. Sheet- and plate-type parts exhibit greater deflection per given load than do structural shapes of beam or column proportions; however, the latter are subject to more deflection than are large cast or forged components. In the design of military vehicles, weight is an important consideration; and the lightest practical construction is used based on

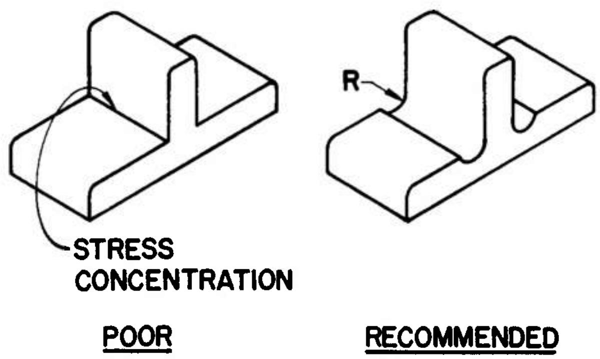


Figure 4-44. Stress Concentration in Castings

load analysis, but augmented by special needs (e.g., armor protection). For structural efficiency, extensive use of sheet-plate or skin-type components is dictated. This construction can result in sizable deflections, depending upon the loads. The designer must provide an optimum number of stiffeners in the structure, positioned to distribute the loads and to hold deflections within allowable limits.

It is important to recognize that different deflection limits rationally exist for different parts of an automotive structure, and even for similar parts under diverse applications. A case in point is a cargo floor. For general duty, the cargo floor is designed primarily for strength, but it is indirectly limited to some elastic deformation when the spacing of its supports is determined. See the example problem in par. 4-2.3.2 for deflection consideration in a particular cargo floor. For heavy duty, a cargo floor is designed entirely for strength; permanent local deformations can and do occur, as when large angular rock cargo is dropped into a dump body. In such cases, local permanent deformations are acceptable[61], and efforts to prevent them would unnecessarily penalize the design.

Methods used to reduce deflections in automotive bodies and hulls are generally similar to those used for the reduction of load concentrations (par. 4-3.2). Obviously, the stiffer structural members in a given construction, and loading surfaces stiffened locally by using heavier material gages or additional layers are useful against deflections as well as against concentrated loads. However, in structural parts that require maximum reduction of deflection, it is the particular positioning of supports with respect to loads that determines good design. Supports, as considered here, can be within the structure itself, or overall vehicle reactions (i.e., from ground-contacting elements and pintle-lunette forces). In massive hulls, not having framelike stiffness, deformations have to be resisted in the cast or plate skins directly.

Every load is accompanied by deflection, but cargo loads have a special relation to deflection of structures. Cargo loads significantly influence vehicle ground reactions both statically and dynamically. Cargo loads, especially of high density, enjoy considerable freedom of position within a load space (see par. 2-7.2 for typical cargo bed sizes of wheeled vehicles). Transverse load magnitude and position are the controlling deflection parameters in a structural member of given material and cross section. Therefore, cargo loads have preferred position with respect to vehicle supports as shown in Fig. 4-46. Effective cargo load distribution to reduce deflection calls for the location of cargo force resultants W_C as near as possible to the main suspension reactions (axles) R_1, on wheeled vehicles ((A), (C), and (D), Fig. 4-46)). Cargo force resultant location on tracked vehicles is not nearly as sensitive (Fig. 4-46(B) and (E)) as for wheeled types because of improved ground pressure distribution $P(x,y)$; but W_C should be positioned near the center of track-ground contact. Preferred cargo position is often impossible or neglected in the military environment. To offset such conditions, loading schedules should be instituted or extended for cargo vehicles. A schedule, permanently displayed on a vehicle, could list simple preferred loading data useful for improved balance and decreased load concentration along with deflection reduction. Similar procedures are employed with aircraft and, to an extent, on land vehicles (e. g., specification of lifting points). Illustration of the importance of cargo

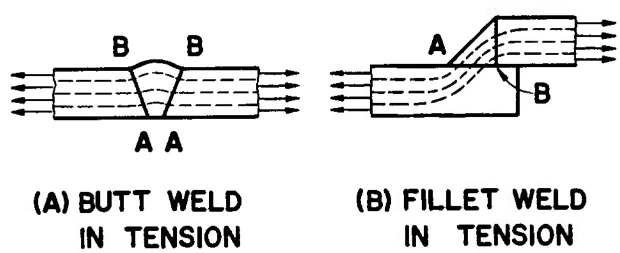


Figure 4-45. Stress Concentration in Welds

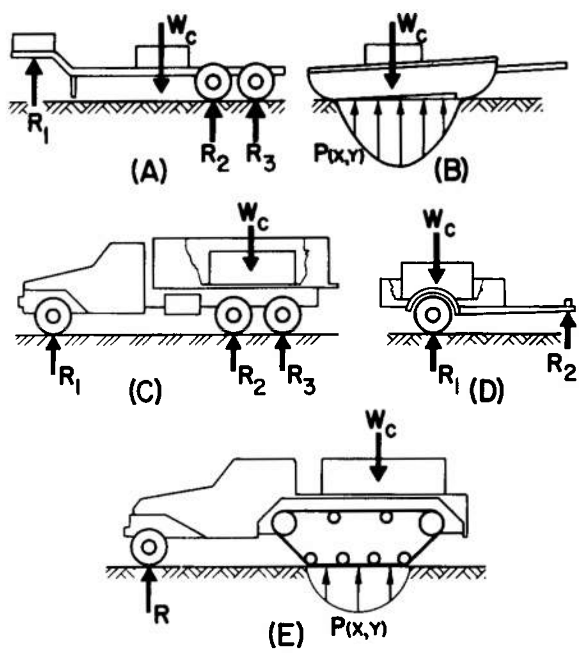


Figure 4-46. Cargo Weight Force Resultants

positioning on deflections is given by the sample problems that follow.

Example 3. Beam type elements.

Consider a vehicle underfloor in which longitudinal members are supported at intervals L = 10 ft (120 in.) and loaded by a concentrated cargo load of W_C, lb. Compare the deflection effects of (A) a preferred cargo position at the quarter-point between supports with (B) the least preferred position located at the midspan. Assume the beam components are fixed supported.

Maximum deflection for condition (A) will be in the longer section of the beam between the load and the support

$$\delta_{(max)_A} = \frac{2W_C a^2 b^3}{3EI\,(L + 2b)^2}\text{, in.} \tag{4-62}$$

Maximum deflection for condition (B) will be under the load

$$\delta_{(max)_B} = \frac{W_C L^3}{192EI}\text{, in.} \tag{4-63}$$

a = the shorter length of beam between load and support = 30 in.

b = the longer length of beam between load and support = 90 in.

E = modulus of elasticity of beam material, lb/in.^{2}

I = moment of inertia of beam cross section, in.^{4}

L = length of span between supports = 120 in.

then

$$\frac{\delta_{(max)_A}}{\delta_{(max)_B}} = \frac{128a^2b^3}{L^3\,(L + 2b)^2} = \frac{128(30)^2(90)^3}{(120)^3(120 + 180)^2} = 0.54$$

Therefore, deflection is reduced by nearly one-half in shifting load location toward a support. It should be noted that reactive loads at supports also change when the applied load position is altered. This is another reason for attempting to place a cargo force resultant near a main support, i.e., a suspension reaction point.

Example 4. Plate-type Elements

The floor in Example 3 is topped with 1/8-in.-thick aluminum plate and loaded by a uniformly concentrated load of q lb/in.^{2} The intervals of lateral support, b = 10 ft, are set by Example 3 but compare the effects of changing longitudinal support spacing—(A) Let a = 3 ft, (B) let a = 6 ft. Assume the plate edges are simply supported.

Maximum deflection for both conditions is at the midpoint of a plate, say

$$\delta = \frac{Cqa^4}{D}\text{, in. } \left[\text{compare to Eq. 4-36, par. 4-2.3.2}\right] \tag{4-64}$$

where

a = shorter side of the plate between supports, in.

$D = \dfrac{Et^3}{12(1 - \mu^2)}$, flexural rigidity for a flat isotropic plate, in.-lb

t = plate thickness, in.

μ = Poisson's ratio, dimensionless

C = deflection coefficient, dimensionless (if plate aspect ratio b/a is substituted for b/L in Fig. 4-20, the values of C_A and C_B can be read directly from the curve C_5)

Thus,

$$\frac{\delta_A}{\delta_B} = \frac{\dfrac{C_A q a_A{}^4}{D}}{\dfrac{C_B q a_B{}^4}{D}} = \frac{C_A a_A^4}{C_B a_B^4}$$

$$\frac{b}{a_A} = \frac{10}{3} = 3.33 \qquad \frac{b}{a_B} = \frac{10}{6} = 1.67$$

$$C_A = 0.0123 \qquad C_B = 0.0085$$

$$\frac{\delta_A}{\delta_B} = \frac{0.0123(3)^4}{0.0085(6)^4} = 0.09$$

The magnitude of delfection is increased approximately 11 times by changing the unsupported plate size from 3 X 10 ft to 6 X 10 ft. Degree of edge restraint has a great influence on extent of deflection.

SECTION II—FLOORS—CONSTRUCTION AND DESIGN

4-4 GENERAL CONSIDERATIONS

Floors of military cargo vehicles are flat wherever possible (excluding corrugations, skid strips or the like) and set as low to the ground as permitted by undulations of terrain that the vehicle is designed to traverse. Some tracked vehicles have the capability of elevating their hulls to clear terrain obstacles. A low floor means lower overall silhouette which is of primary concern in combat vehicles. Tracked vehicles have their floor level below the tops of tracks, requiring roughly a T-shaped hull cross section due to track sponsons. Lower operating height is traded off against reduced hull volume. In wheeled vehicles the entire floor may set above the wheels (plus clearances for loading and suspension system deflections) or the primary floor may be set below them (Fig. 4-47(A)), reducing cargo space due to obstruction from the wheel housings. An unobstructed floor (Fig. 4-47(B)) permits hauling of wider cargo, but with sacrifice of loading height and operating silhouette. Fig. 4-47 compares loading height H and cargo width W between these two floor arrangements for otherwise identical size wheeled vehicles and equal height of cargo h. In some designs, provision is made for converting an interrupted floor to a completely level floor by means of portable floor plates for certain cargo sizes or vehicle missions.

4-4.1 TYPES OF FLOORS

To a large extent, vehicle type dictates the kind of floor construction encountered in military automotive practice. In tracked vehicles, portions of large hull castings or plate fabrications form the floors as shown in Figs. 4-48 and 4-49, respectively. Often, the actual cargo, equipment, or personnel floor will be higher than the hull floor itself because of obstructions at the bottom of the hull. In such cases, additional floor plates are mounted on supports provided in the primary floor or walls. This auxiliary floor can be of quite light construction for personnel and many equipment transport applications. Common obstructions requiring a raised floor are torsion bar suspension springs and their protective housings which penetrate transversely into the hull, as shown in Fig. 4-49. Little space is used by the torsion bars and the secondary floor can be located close to the primary floor. Fig. 4-50(A) and (B) shows before and after views in placement of the secondary floor over the longitudinal centerline well in an amphibious tracked vehicle. In this example, the power plant as well as torsion bars are buried in the center well. In wheeled vehicles, the chassis-body

In wheeled vehicles, the chassis-body relationships (par. 1-4) mainly determine the type of floor construction. Separate frame and body permit the simplest construction of floors as well as other body parts. Here, the floor is a load-bearing covering over a normally transversely stiffened platform which, in turn, is mounted on longitudinal frame members. Fig. 4-51 clearly illustrates this construction principle for a particular body type (platform) and floor covering, but it must be emphasized that floors of other body types (par. 1-5) or other floor coverings are assembled in the same general manner. The integral frame and body design of wheeled vehicles calls for a fabricated combination of structural shapes, formed and stamped sheet metal sections, plates, etc. Integration of the various components into a unit floor is accomplished by bolting, riveting, and welding, the latter especially in amphibious

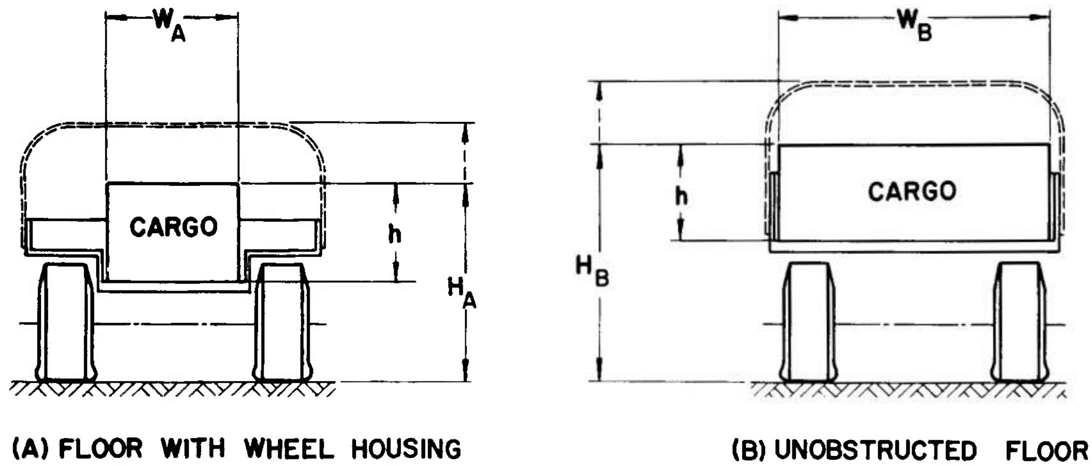


Figure 4-47. Floor Cross Section Effects on Cargo Width and Loading Height

vehicles. Use of sandwich panels (honeycomb) for flooring as well as other body parts and bonding for the fabrication method is well into the developmental stage. The unitized floor is the actual payload surface in certain vehicles; other wheeled vehicles, notably those reflecting increased emphasis on amphibian capabilities, carry the integral construction to hull-like bodies which closely resemble hulls of amphibious tracked vehicles. Construction features of tracked or wheeled amphibious vehicles will be similar except for extra material thickness requirements of combat (as opposed to tactical) types; therefore, discussion of primary and secondary floors for tracked amphibious vehicles, earlier in this subparagraph, applies to wheeled varieties also. Advantages and disadvantages of separate and integral frame-body relationships as a whole, are listed in par. 1-4; most of these relative advantages and disadvantages hold for floor considerations.

The payload surface itself can be one of several types, each with certain favorable and unfavorable characteristics.

a. *Sheets and plates.* Flat shapes offer ease of installation, cleaning, and production economy; corrugated shapes have higher strength for the same weight, making them useful for concentrated or impact loads, but repairs and cleaning are more involved than with flat sheet or plate. Standard corrugated profiles are circular and trapezoidal, with pitch, trough depth, and symmetry as additional variables. Flat plate, available in several metals with several raised tread patterns for safety purposes, finds use in vehicles where sure-footedness in entering, exiting, or moving about is essential, i.e., on personnel carriers, wreckers, and certain cargo vehicles. Coatings of nonskid materials are sometimes applied to such surfaces. Advantages of high strength-to-weight ratio sheets and plates are offset by their higher cost.

b. *Extruded thin-wall section planks.* Both flat-top and corrugated are available in light metal (mainly aluminum) in very many cross sections of interest to the vehicle designer. Typical planks have small width but great maximum length that can be suited to specific

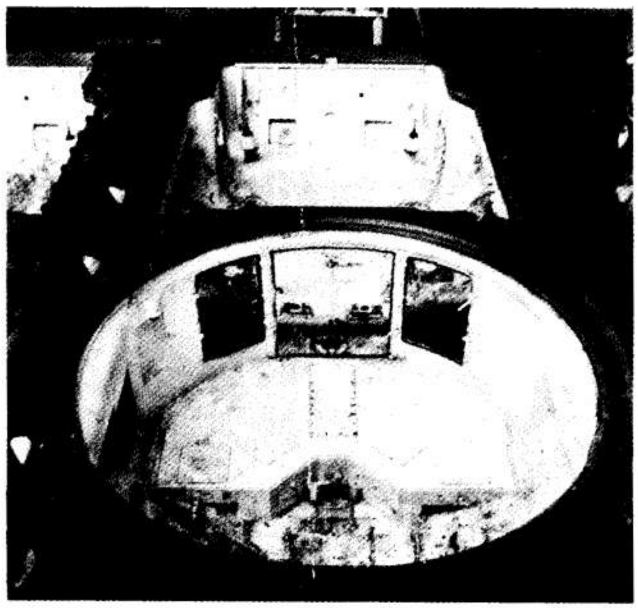


Figure 4-48. Typical Floor and Wall Construction, M48 Tank

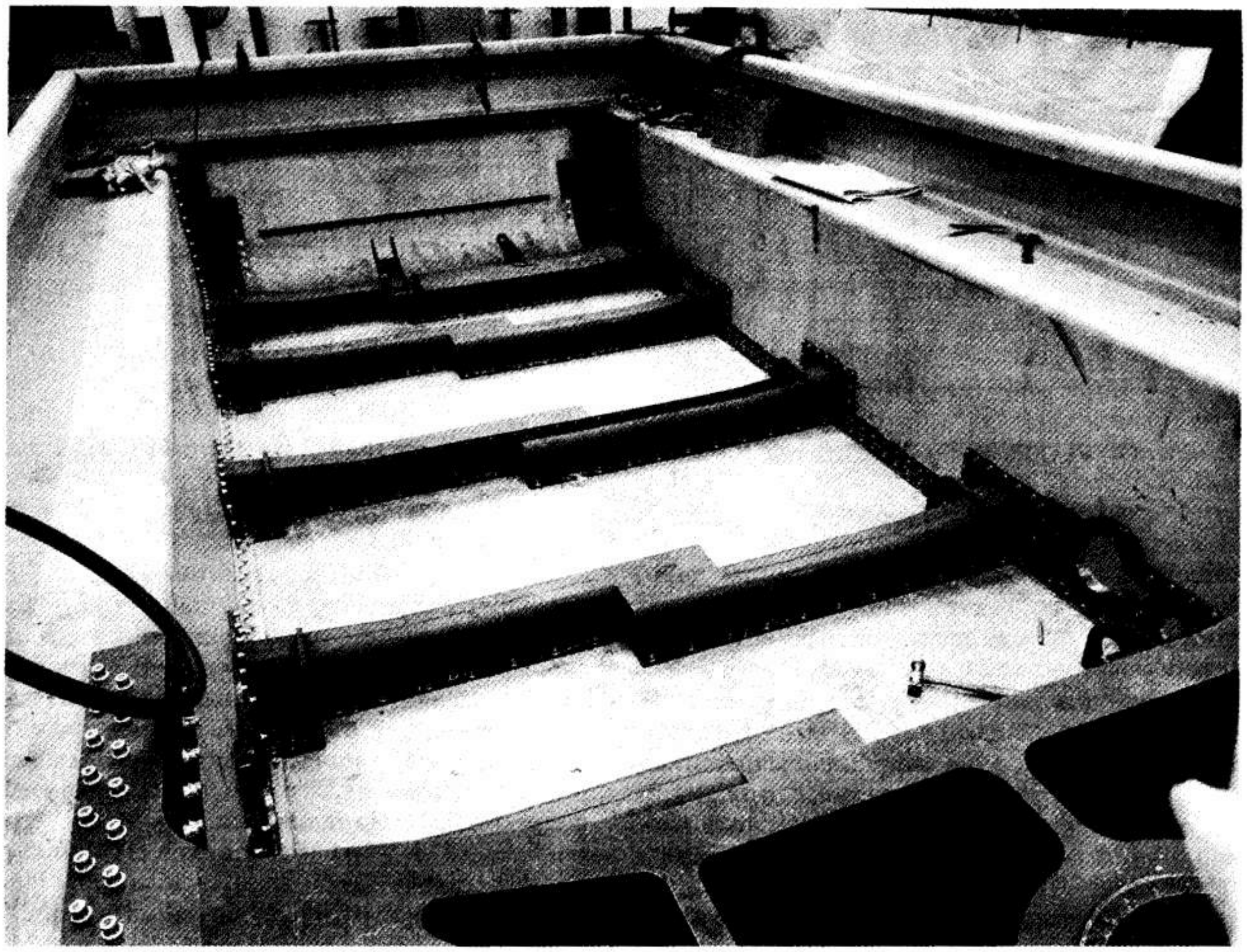
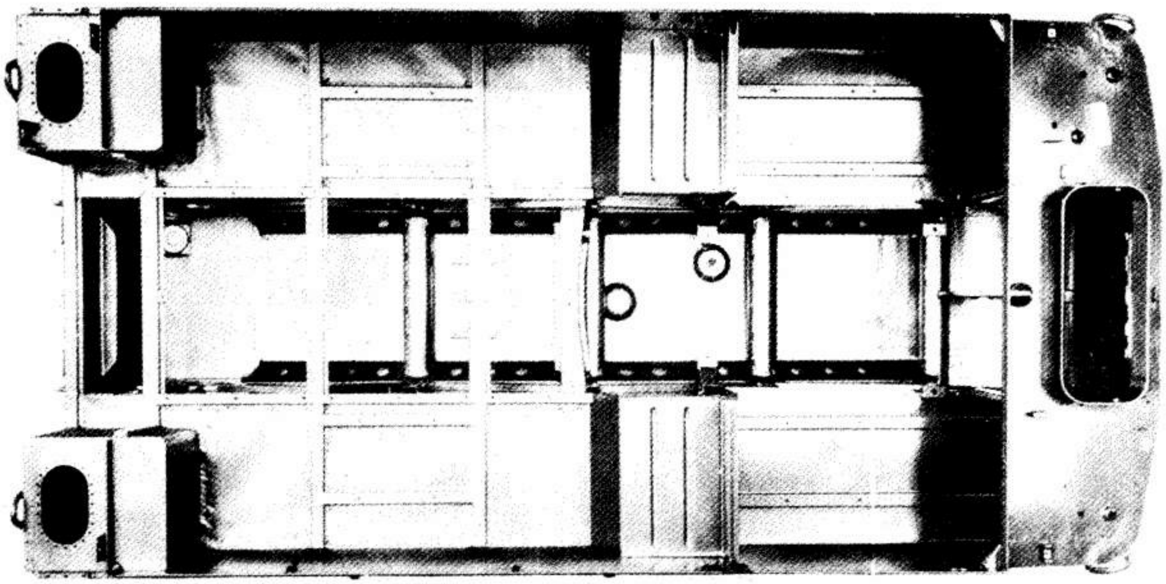


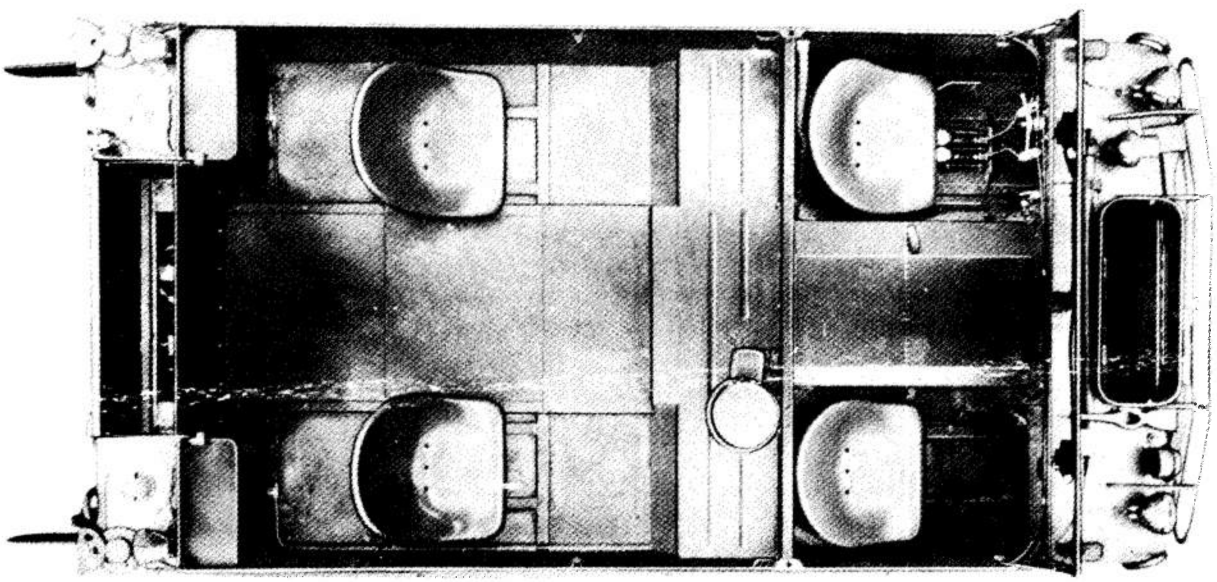
Figure 4-49. Typical Floor and Wall Construction, XM138 Self-propelled Howitzer (Suspension System Torsion Bar Spring Housings Are Also Shown)

needs. Their efficiency comes into play mainly in separate frame and body vehicles where they are conveniently mounted to transverse floor supports. Fig. 4-51 shows this flooring type using corrugated planks. Further advantage of planking is that novel interconnecting methods can be designed into the cross sections or special interlocking components used to assemble complete floors economically and with minimum fasteners. Fig. 4-52(A) to (D) shows corrugated and nonflat-top planks of typical shape and dimensions used for flooring in cargo trucks and trailers. The deeper corrugated shapes in these examples are not so much used for increased strength, but rather for improved air circulation around cargo and end draining of collected moisture encountered in refrigerated vehicles. Fig. 4-52(E) gives transverse cross sections of complete floor systems. Four representative examples of flat-top floor planks with wide automotive applications appear in Fig. 4-53; example (D) is particularly suited for a treadplate or catwalk. Important shape selection parameters discussed in par. 4-2.2 are also listed in Fig. 4-53. Illustrations of plank interconnecting methods and techniques of mounting them to floor supports are given in Fig. 4-54; methods of parts (A), (B), and (C) are usable with corresponding marked planks of the previous figure. Usual mounting methods (Fig. 4-54) necessitate drilling through plank and support or at least the support, if bolt head retaining sections are used. If drilling is impractical or if supports are unduly weakened thereby, mounting may be accomplished as suggested in Fig. 4-54 via dip sections and bolt adapters of Fig. 4-54(B) and (C).

Plank-type flooring can be oriented both longitudinally and transversely with respect to a vehicle, although the former system is more



(A) Top View of Hull



(B) Top View of Hull Assembly Less Canvas Top

Figure 4-50. Typical Floor Construction, T107 Amphibious Cargo Carrier

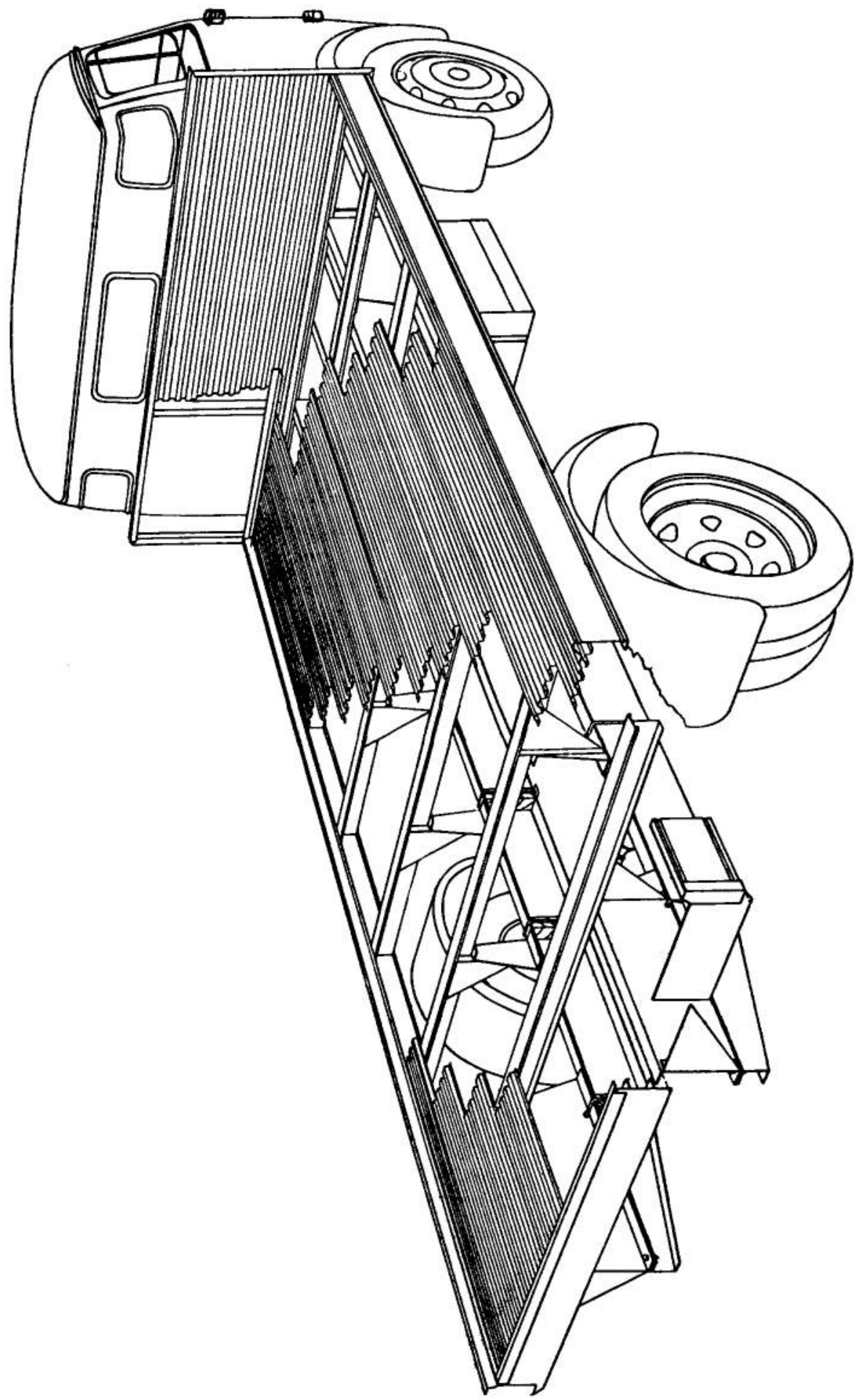


Figure 4-51. Typical Floor Construction for a Vehicle With Separate Frame and Body[71]

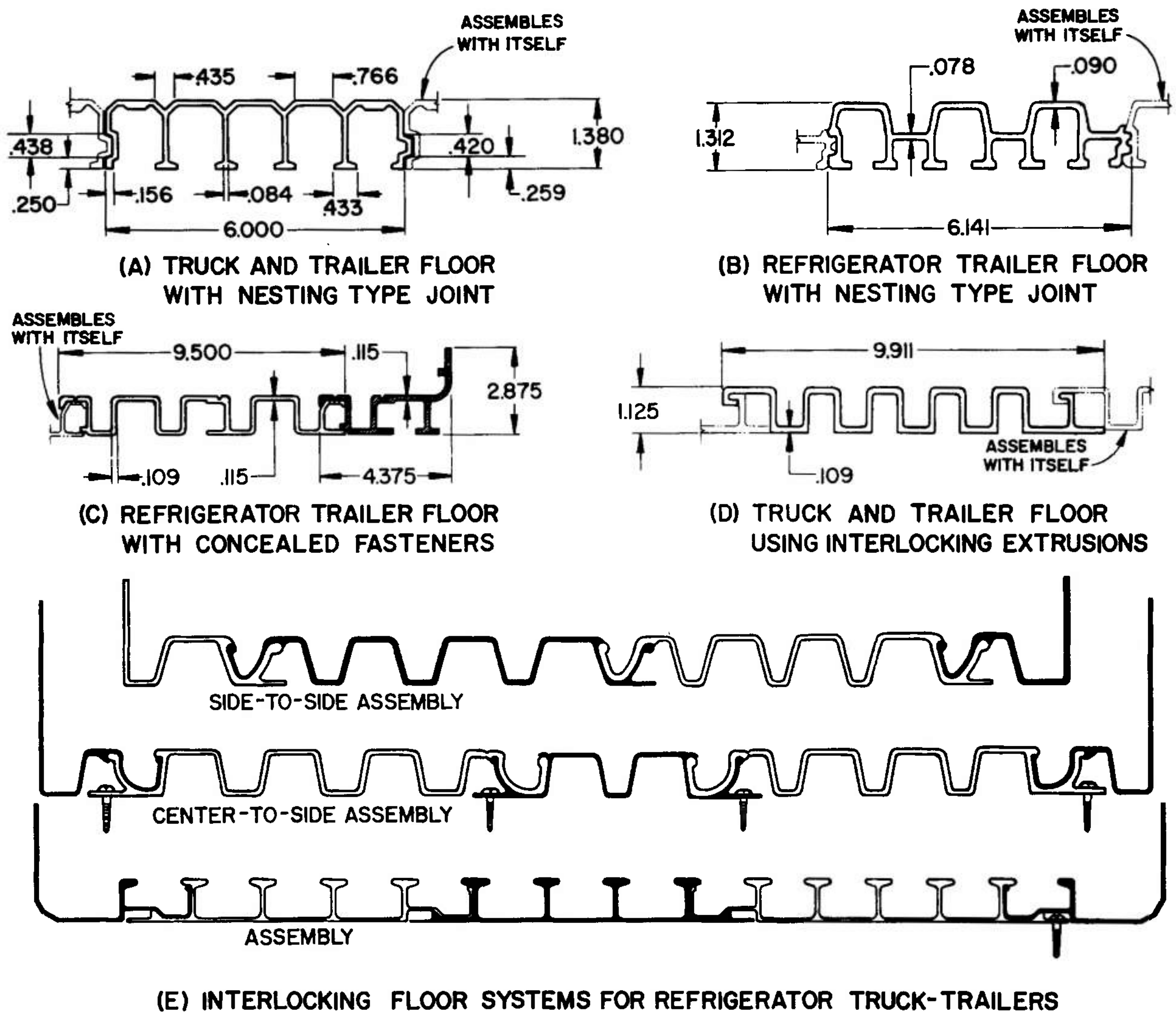


Figure 4-52. Typical Corrugated and Nonflat-top Aluminum Plank Flooring[72]

widely used[71]; such directional considerations have no meaning for plate-type flooring. Transverse flooring, in turn, requires use of longitudinal supports or stiffeners, but offers advantage through smaller, but more numerous, parts. For the military environment this has potential in better supply, storage, economy, and ease of maintenance. Transverse considerations are limited to flat-top flooring because corrugations running in that direction could impede fore-aft cargo handling flow as well as present problems for vehicle cleaning.

c. *Sandwich panels.* Floors can be constructed of few components using the efficiency of sandwich construction, which is still under development for land vehicles. Par. 4-2.3.2 gives substantial information on the state-of-the-art.

d. *Wood flooring.* Use of wood for primary or permanent floor construction is found in the rather light load environment of vans, particularly of the office-command post or electronic-instrumentation type (par. 1-5.2). This does not imply that wood is not used for more rugged applications. In heavier load situations wood is utilized as a secondary, load-cushioning floor, normally sandwiched between metal primary and bottom floors (Tables 4-9 and 4-10). Still another application of wood as a vehicle load-bearing element is the expendable top flooring found on a variety of land vehicles. Unusually heavy or concentrated loads encountered in hauling complete vehicles

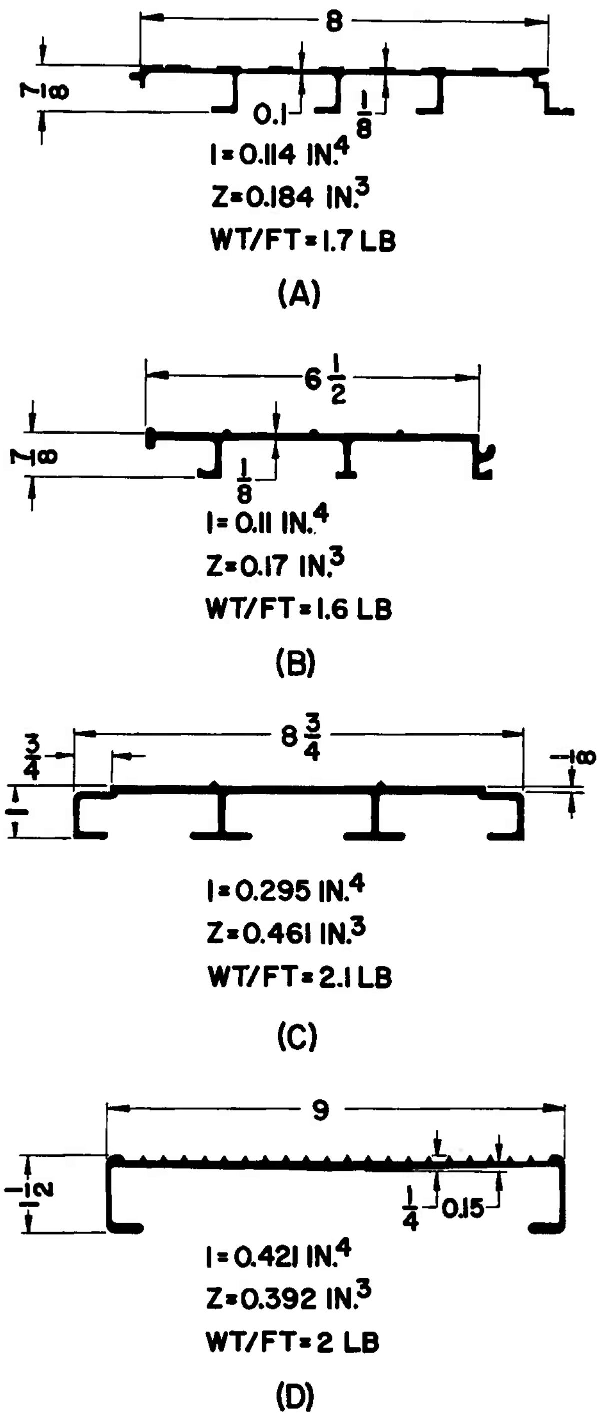


Figure 4-53. Typical Aluminum Flat-top Plank Flooring[73]

and machinery are absorbed with thick wood flooring or local pads of this type. Wood is well-suited as an expendable floor material, but it has economic disadvantages for permanent installations, because of special mounting, joining, and preservation requirements associated with it. MIL-W-3912A[74] covers fabricated wood parts applicable to automotive bodies.

Military Specifications of particular vehicles contain valuable information on body construction details. Minimum material thicknesses of floors (and other body members) are to be found in many of the specifications. There are large numbers of such documents and specific thickness requirements can be expected to vary greatly with type and size of vehicle. Some typical minimum floor thicknesses based on randomly selected Military Specifications are offered in Tables 4-9 and 4-10 as rough guidance to the designer. Table 4-10 gives other body thicknesses also. On a particular design task, it is recommended that several specifications be obtained for the type of vehicle under consideration to provide much needed information on existing construction procedures and details. Comprehensive subject lists of Military Specifications are available for this purpose. Thicknesses shown in the tables are exclusive of subframe or under-floor support members.

4-4.2 SKID STRIPS

An effective method of protecting floor surfaces from damage and wear of applied loads is to increase floor thickness locally with skid strips and wear bars. Such a provision also

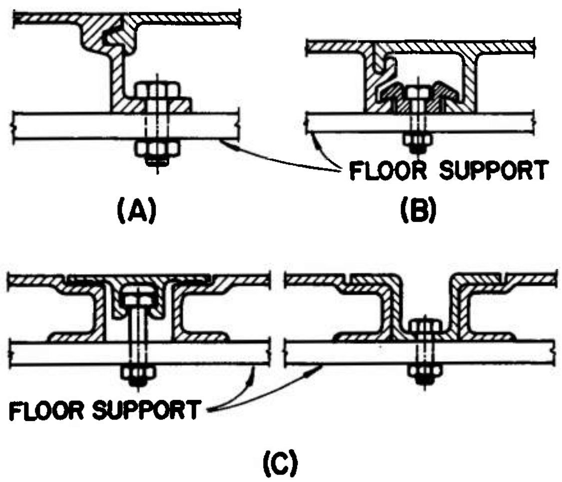


Figure 4-54. Typical Mounting and Interlocking Methods for Flat-top Plank Flooring[73] *(Methods (A), (B), and (C) are usable with correspondingly marked planks of Fig. 4-53.)*

TABLE 4-9 TYPICAL VEHICLE FLOOR THICKNESSES

Body Type	Vehicle Size	Construction	Minimum Thickness, in.	Gage*	Reference
Military Cargo	5-ton 6 X 6 Series M39 Truck	Floor pan (sheet steel)	0.0209	25	MIL-T-740A
		Floor boards (lumber)	1-5/16	—	
		Wear plate (sheet (steel)	0.1046	12	
		TOTAL FLOOR	1-7/16	—	
Van		Floor pan (steel)	0.0179	26	
		Floor boards (lumber)	1-1/8	---	
		TOTAL FLOOR	~1-9/64		
Dump	17-ton GVW 4 X 4 truck	Floor (sheet steel)	0.1644	8	MIL-T-26150D (USAF)
	12-ton GVW High lift 4 X 2 commerciel truck				MIL-T-46797 (MO)

*Manufacturers' standard gage for steel

TABLE 4-10 TYPICAL STEEL DUMP BODY THICKNESSES

Vehicle Rated Minimum GVW, in.	Minimum Material Thickness, in.				
	Body Type	Cab Protector	Front Plates	Floor Plates	Side Plates
40,000	Double bottom	1/4	3/8	3/8 wear, 1-1/2 wood, 1/4 bottom	3/8
61,000	Double bottom	1/4	3/8	3/8 wear, 1-1/2 wood, 1/4 bottom	1/2
70,000	Double bottom	1/4	3/8	1/2 wear, 1/4 bottom	1/2
	Alloy steel*	1/4	3/8	3/8 wear, 1/4 bottom	3/8
86,500	Double bottom	1/4	3/8	1/2 wear, 3/8 bottom	1/2
	Alloy steel*	1/4	3/8	1/2 wear	3/8
91,000	Double bottom	1/4	3/8	1/2 wear, 1/4 bottom	1/2
	Alloy steel*	3/16	3/8	3/4 wear	3/8
103,000	Double bottom	1/4	3/8	1/2 wear, 2 wood, 3/8 bottom	1/2
	Alloy steel*	3/16	3/8	5/8 wear	5/18
150,000	Double bottom	3/16	1/2	5/8 wear, 3/8 bottom	3/8
	Alloy steel*	1/4	3/8	5/8 wear	5/16

*Alloy steel thickness minimums are based on material having not less than 95,000 psi yield strength.

facilitates cargo loading and unloading in a direction along the strips or bars. Strictly speaking, both items pertain to metal overlays on flat-top floors, since corrugated and contoured floors (par. 4-4.1) have "built-in" skid strips. Skid strips are sheet metal strips of either flat or formed shallow cross sections that are riveted or welded to floors and become the primary loading surfaces. Strips are placed at regular intervals along the floor with their orientation usually parallel to the longitudinal centerline of the vehicle. Such an arrangement aids arrangement aids fore-aft cargo flow. Transverse orientation of strips is also feasible and potentially usable for newer concept cargo vehicles with side loading, but the drawbacks mentioned for transversely corrugated floors again apply.

Although the traditional skid strip is widely used, its extra parts and labor requirements do not represent efficient design. For vehicles that can make use of extruded planks (par. 4-4.1), this flooring type offers economy and efficiency from the standpoint of skid or wear strips also. Fig. 4-54 shows some examples of skid strips made integrally with floor planks. Obviously, heavier strip patterns, especially of style (A), can be produced with modified extrusion dies. Corrugated and nonflat-top planks (Fig. 4-52) also have members that inherently act as skid strips, but such flooring often finds special use in inclosed refrigerated vehicles (Fig. 4-53 and par. 4-6.1) wherein flooring contours are utilized to aid refrigeration rather than primarily provide for strength and wear.

Severity of floor loading indicates the kind and size of skid strip or wear surface needed on a vehicle. For most vehicles and cargo types, the sheet metal overlay or the integral skid strip is satisfactory. However, for very severe loading conditions, the heavier wear bar and wear plate replace the skid strip. Use of wearing surfaces becomes increasingly important on aluminum bodies because of wear resistance inferior to that of steel. Dump bodies (par. 1-5.3) hauling bulk construction materials, especially rock and other abrasive materials in off-highway runs, well qualify for severe wear, overload, and impact environment. On such bodies, two kinds of wear surfaces are used to protect floors and floor-to-side corners—(1) Wear bars for hard materials containing sizable rock, and (2) wear liner plates for materials containing small sharp rock particles. Fig. 4-55 illustrates the construction of several heavy-duty wear surface installations on dump bodies along with some important body dimensions. The examples pertain to aluminum bodies with steel wear bars and plates attached by bolting or welding. In one installation, aluminum spacers are used between wear bars and the main floor. Actually, aluminum wear surfaces can also be used, but

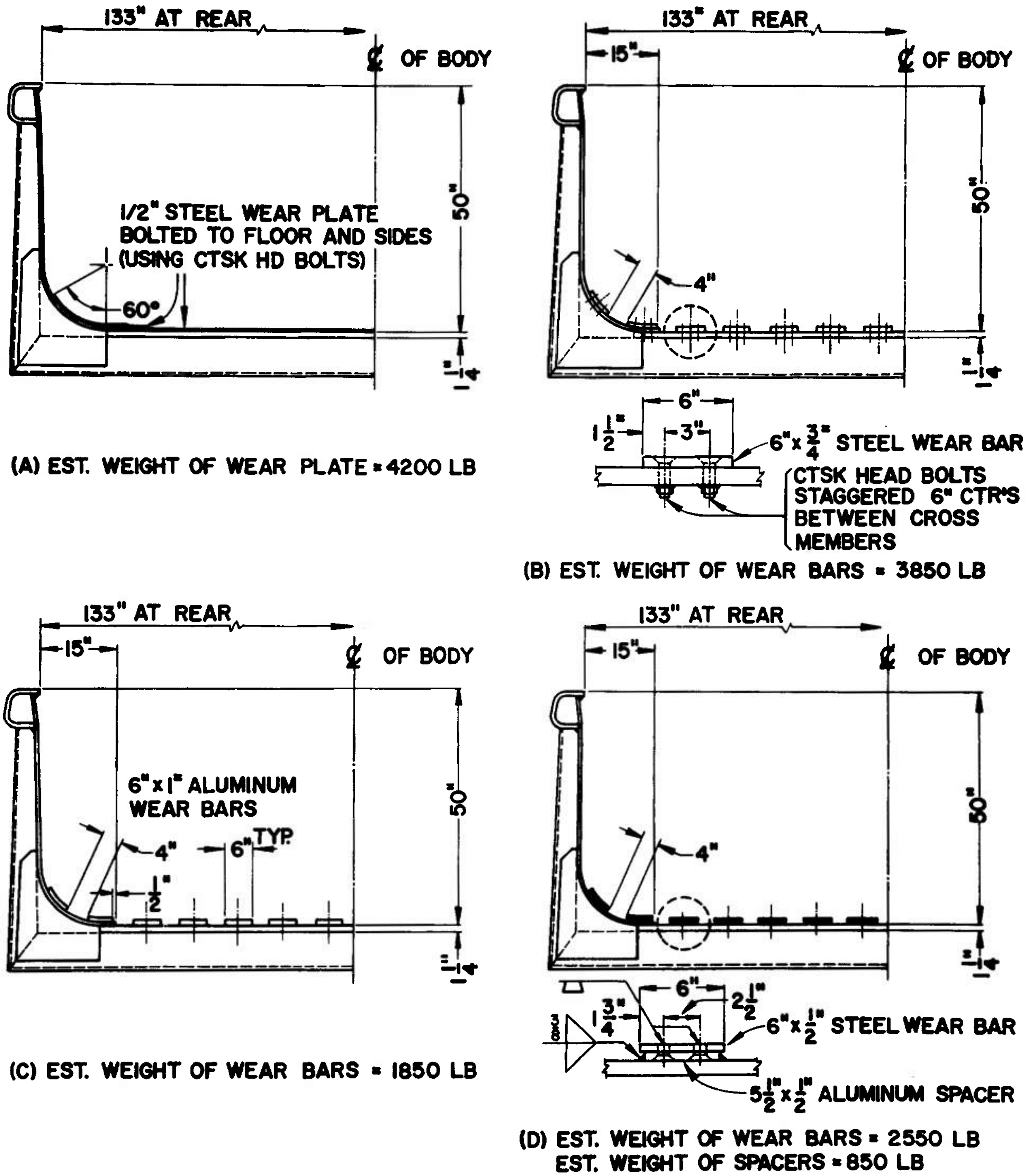


Figure 4-55. Heavy-duty Wear Surface Installations on Dump Bodies[75]

these would require more frequent replacement than similar steel wear elements[75].

4-4.3 WATERTIGHTNESS IN FLOORS

As considered here, watertightness deals with the design and construction of floors to prevent entrance of water through the undersides of automotive bodies and hulls during water-crossing operations. A lesser problem (to military vehicles at any rate) concerns internal watertightness—necessary for moisture collection, draining, and cleaning procedures associated with refrigerated cargo. The existence of this problem, is, however, mentioned.

Watertightness in military land vehicles is strictly a function of a construction method which, in turn, is determined by the mission a vehicle is designed to perform. Various degrees of water-crossing ability are defined in Refs. 76 and 77, ranging from a very minimal fording (nonamphibious) to rigorous full amphibious requirements. Ref. 78 contains a brief, pertinent discussion of amphibious vehicle categories. While water-crossing requirements are aimed mainly at future military vehicles, certain minimum abilities are set forth for standard vehicles through practicable modifications. The less demanding water-crossings are accomplished by using a combination of waterproofing, fording kits, and flotation kits (par. 3-36) on standard, separate frame and body wheeled vehicles; similar methods, plus the inherent watertight (but not sufficiently self-buoyant) construction of the hull, are used on tanks.

The importance of watertight floors has increased with the standardization of full amphibious, tracked vehicles, and, more recently, with demands and emphasis on wheeled vehicles to swim on inland waterways, at least, with full payload. Vehicles meeting this capability have a unitized body or hull structure (par. 1-4.2) with all welded construction. The type of welding specified, complemented by water tests, is the greatest single feature that assures watertight integrity. The undersides of amphibious automotive structures are largely sealed surfaces that do not present leakage problems; it is the access openings that are troublesome. On wheeled vehicles such openings are needed to allow driveline components to pass through the body. Openings for these purposes can be made generally small, which combined with proper seal design has been successful in the production of watertight vehicular floors. Floors of track-laying vehicles have similar problems, but there are more numerous driveline access openings than on wheeled vehicles and hull bottom escape hatches, in some cases, to complicate the task of sealing, as shown in Fig. 3-8. Seal design for these applications is discussed in Section III, Chapter 3. The greater problem of watertightness occurs in floor-to-wall joints and at the extensive body-side panel openings of wheeled vehicles (par. 4-6.2).

4-5 LOADS ON FLOORS

4-5.1 CARGO LOADS ON FLOORS

Distributed and concentrated are the broadest categories into which cargo loads can be classed. Examination of several typical cargo forms, however, reveals that distributed rather than truly concentrated loads well describe most cargo forces, either through natural occurrence or special load-distribution techniques (par. 4-3). Clear understanding of the word "distributed" is important. It does not imply uniform distribution, since heaped granular cargo can have linear or parabolic distribution; more irregular load distributions are also possible. However, uniform distribution of cargo loads is commonplace, as can be verified by some examples. Hydrostatic floor loads (par. 4-5.3) and fluid cargo of tank-type bodies (par. 4-16) certainly exhibit uniformly distributed loading. Bulk, crated, and boxed cargo quite often have uniform density or at least a nominal density range (Table 4-8). Normally, uniform loading height is observed for such cargo, which leads to uniform weight distribution. Containerization of cargo—crates, boxes, pallets, drums, cans—makes cargo height standardization convenient. Loose material of granular or particle composition can act as either a uniformly or nonuniformly distributed load; this is determined by the extent to which a transporting vehicle is loaded. The Society of Automotive Engineers[80] defines and recommends loading capacities for bodies that haul these types of cargo. *Struck capacity* V_S is defined as the volume enclosed by the body proper to the level of (or "struck off" at) the top of the side walls and specified to the nearest 0.1 yd^3. *Heaped capacity* V_H is defined

TABLE 4-11 APPROXIMATE DENSITIES OF COMMON MATERIALS[79]

Substance	*Average Density,* lb/ft^3	*Substance*	*Average Density,* lb/ft^3
Metals, Alloys, Ores		Cotton, flax, hemp	93
Aluminum,		Fats	58
cast-hammered	165	Flour, loose	28
Aluminum, bronze	481	Flour, pressed	47
Brass, cast-rolled	534	Glass, common	162
Bronze, 7.9% to 14% Sn	509	Glass, plate or crown	161
Bronze, phosphor	554	Glass, crystal	184
Copper, cast-rolled	556	Glass, flint	247
Copper, ore, pyrites	262	Hay and straw, bales	20
German silver	536	Leather	59
Gold, cast-hammered	1205	Paper	58
Gold coin (U. S.)	1073	Potatoes, piled	44
Iridium	1383	Rubber, caoutchouc	59
Iron, gray cast	442	Rubber goods	94
Iron, cast, pig	450	Salt, granulated, piled	48
Iron, wrought	485	Saltpeter	132
Iron, spiegel-eisen	468	Starch	96
Iron, ferrosilicon	437	Sulphur	125
Iron ore, hematite	325	Wool	82
Iron ore, limonite	237		
Iron ore, magnetite	315	*Timber, Air-dry*	
Iron slag	172	Apple	44
Lead	710	Ash, black	34
Lead ore, galena	465	Ash, white	42
Manganese	475	Birch, sweet, yellow	44
Manganese ore, pyrolusite	259	Cedar, white, red	42
Mercury	847	Cherry, wild red	27
Monel metal, rolled	555	Chestnut	30
Nickel	537	Cypress	29
Platinum, cast-hammered	1330	Fir, Douglas	32
Silver, cast-hammered	656	Fir, balsam	25
Steel, cold-drawn	489	Elm, white	35
Steel, machine	487	Hemlock	29
Steel, tool	481	Hickory	48
Tin, cast-hammered	459	Locust	45
Tin ore, cassiterite	418	Mahogany	44
Tungsten	1200	Maple, sugar	43
Zinc, cast-rolled	440	Maple, white	33
Zinc, ore, blende	253	Oak, chestnut	46
		Oak, live	54
Various Solids		Oak, red, black	42
Cereals, oats, bulk	26	Oak, white	48
Cereals, barley, bulk	39	Pine, Oregon	32
Cereals, corn, rye, bulk	45	Pine, red	30
Cereals, wheat, bulk	48	Pine, white	27
Cork	15	Pine, Southern	38-42

TABLE 4-11 APPROXIMATE DENSITIES OF COMMON MATERIALS[79] (Cont'd)

Substance	*Average Density,* lb/ft³	*Substance*	*Average Density,* lb/ft³
Pine, Norway	34	Sandstone, bluestone	110
Poplar	27		
Redwood, California	26	*Brick Masonry*	
Spruce, white, red	28		
Teak, African	62	Hard brick	128
Teak, Indian	48	Medium brick	112
Walnut, black	37	Soft brick	103
Willow	28	Sand-lime brick	112
Various Liquids		*Concrete Masonry*	
Alcohol, ethyl, 100%	49	Cement, stone, sand	144
Alcohol, methyl, 100%	50	Cement, slag, etc.	130
Acid, muriatic, 40%	75	Cement, cinder, etc.	100
Acid, nitric, 91%	94	*Various Building Materials*	
Acid, sulphuric, 87%	112		
Chloroform	95	Ashes, cinders	40-45
Ether	46	Cement, portland, loose	94
Lye, soda, 66%	106	Portland cement	196
Oils, vegetable	58	Lime, gypsum, loose	53-64
Oils, mineral, lubricants	57	Mortar, lime, set	103
Turpentine	54		94
Water, 4 °C, max density	62.428	Mortar, portland cement	135
Water, 100 °C	59.830	Slags, bank slag	67-72
Water, ice	56	Slags, bank screenings	98-117
Water, snow, fresh fallen	8	Slags, machine slag	96
Water, sea water	64	Slags, slag sand	49-55
		Earth, etc., Excavated	
Ashlar Masonry			
Granite, syenite, gneiss	159	Clay, dry	63
Limestone	153	Clay, damp, plastic	110
Marble	162	Clay and gravel, dry	100
Sandstone	143	Earth, dry, loose	76
Bluestone	153	Earth, dry, packed	95
		Earth, moist, loose	78
		Earth, moist, packed	96
Rubble Masonry			
		Earth, mud, flowing	108
Granite, syenite, gneiss	153	Earth, mud, packed	115
Limestone	147	Riprap, limestone	80-85
Sandstone	137	Riprap, sandstone	90
Bluestone	147	Riprap, shale	105
Marble	156	Sand, gravel, dry, loose	90-105
Dry Rubble Masonry		Sand, gravel, dry, packed	100-120
Granite, syenite, gneiss	130		
Limestone, marble	125	Sand, gravel, wet	126

TABLE 4-11 APPROXIMATE DENSITIES OF COMMON MATERIALS[79] (Concluded)

Substance	*Average Density,* lb/ft³	*Substance*	*Average Density,* lb/ft³
Excavations in Water		*Bituminous Substances*	
Sand or gravel	60	Asphaltum	81
Sand or gravel and clay	65	Coal, anthracite	97
Clay	80	Coal, bituminous	84
River mud	90	Coal, lignite	78
Soil	70	Coal, peat, turf, dry	47
Stone riprap	65	Coal, charcoal, pine	23
		Coal, charcoal, oak	33
		Coal, coke	75
Minerals		Graphite	135
		Paraffin	56
Asbestos	153	Petroleum	54
Barytes	281	Petroleum, refined (kerosene)	50
Basalt	184		
Bauxite	159	Petroleum, benzine	46
Bluestone	159	Petroleum, gasoline	45
Borax	109	Pitch	69
Chalk	143	Tar, bituminous	75
Clay, marl	137		
Dolomite	181	*Coal and Coke, Piled*	
Feldspar, orthoclase	162		
Gneiss	175	Coal, anthracite	47-58
Granite	165	Coal, bituminous, lignite	40-54
Greenstone, trap	187		
Gypsum, alabaster	159	Coal, peat, turf	20-26
Hornblende	187	Coal, charcoal	10-14
Limestone	155	Coal, coke	23-32
Marble	170		
Magnesite	187		
Phosphate rock, apatite	200		
Porphyry	172		
Pumice, natural	40		
Quartz, flint	165		
Sandstone	143		
Serpentine	171		
Shale, slate	172		
Soapstone, talc	169		
Syenite	165		
Stone, Quarried, Piled			
Basalt, granite, gneiss	96		
Limestone, marble, quartz	95		
Sandstone	82		
Shale	92		
Greenstone, hornblende	107		

as the total of struck capacity plus the volume enclosed by four planes at a slope of one vertical and two horizontal (2:1) extending upward and inward from the top edges of the side and end plates or load-carrying extensions thereof and specified to the nearest 0.5 yd^3 for bodies with $V_S < 12$ yd^3 and to the nearest 1.0 yd^3 for bodies with $V_S > 12$ yd^3. The preceding definitions refer to truck and trailer bodies of standard box configurations (Fig. 4-56). More elaborate dump, scraper, or wagon bodies and other details are covered in Ref. 80. Finally, heavy, odd-shaped metal parts, machinery, and complete vehicles make up a type of cargo with quasidistributed loads; i.e., the weights of objects are distributed at least into feet, pads, supports, and other contact surfaces that normally exist or are artificially supplied by load distribution techniques (par. 4-3). Therefore, most existing floor designs and calculations are based on uniformly distributed cargo loading (true for walls and roofs also).

Table 4-11 lists approximate densities of materials, many of which are military cargo at some time. Although the listed densities are only approximate, the values are useful for computing load magnitudes in applications where a non-complex cargo volume can be established. The troublesome variable connected with cargo volume is height of load. For containerized and bulk or loose material of uniform distribution and density, it presents no problem. For heaped loads of loose cargo, maximum cargo height H can be found by writing an equation, using the SAE definition:

$$H = h_1 + h_2 = h_1 + \frac{w}{4}, \text{ in.} \tag{4-65}$$

where

h_1 = inside height of cargo box sides, in.
h_2 = maximum height of cargo above cargo box sides, in.
w = inside width of cargo box, in.

A dump body loaded to heaped capacity represents a severe cargo load condition. The body is more completely loaded than possible with other cargo forms and, depending on relative densities, this condition can amount to more total load than due to some so-called "heavy" solid cargo.

The SAE-recommended heaped slope and Eq. 4-65 correspond to approximately a 26.6° angle of repose in the material. This is in line with the natural angle of repose found in piled bulk materials, representative values of which range from 27° to 40° (Ref. 78). If there is a need to use a different slope or any linearly varying load height, Eq. 4-65 can be appropriately modified

$$H = h_1 + \frac{kw}{2}, \text{ in.} \tag{4-66}$$

where

k = slope of the heaped material above the cargo box (vertical to horizontal), dimensionless

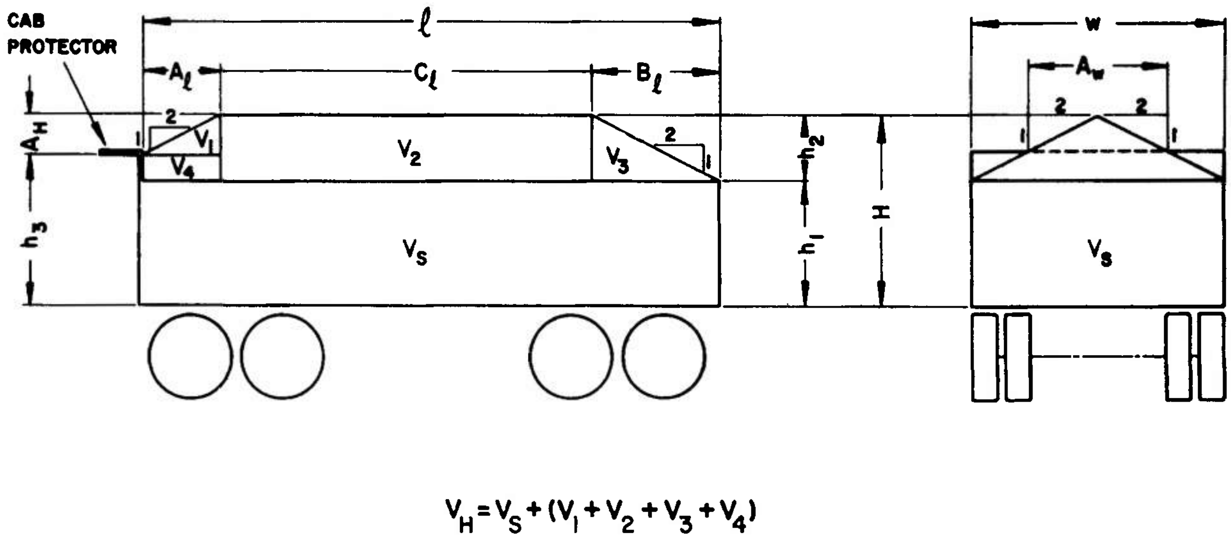


Figure 4-56. Struck Capacity and Heaped Capacity for Box-type Dump Bodies[59]

Cargo load intensity q can then be conveniently expressed for uniform density materials by

$$q = h\rho, \text{lb/in.}^2 \tag{4-67}$$

where

h = cargo height, general (when $h = H$, $q = q_{(max)}$), in.

ρ = material density, lb/in.^{3}

Sample Problem

A 15-ton dump truck of the configuration shown in Fig. 4-56 is to carry damp, plastic clay. Inside dimensions of the body are:

length $\ell = 12$ ft

width $w = 8$ ft

height of side plates $h_1 = 34$ in.

height of cab protector above side plates $h_3 = 18$ in.

Find (a) maximum load height H; (b) maximum load intensity $q_{(max)}$, (c) heaped capacity of the body V_H; and (d) total payload W_P for an assumed 2:1 (horizontal to vertical) slope of heaped loading.

(a) $H = h_1 + \dfrac{w}{4} = 34 + \dfrac{96}{4} = 58$ in. [Eq. 4-65]

(b) density of cargo $\rho = 110$ lb/ft^3 = 0.0637 lb/in.^{3} [Table 4-11]

$q_{(max)} = H\rho = 58(0.0637) = 3.69$ lb/in.^{2} [Eq. 4-67]

(c) Refer to Fig. 4-56:

$$V_1 \begin{cases} \text{height: } A_H = H - (h_1 + h_3) = 58 - 52 \\ \qquad = 6 \text{ in.} = 0.5 \text{ ft} \\ \text{base width: } A_w = 4A_H = 2 \text{ ft} \\ \text{base length: } A_\ell = 2A_H = 1 \text{ ft} \end{cases}$$

$$V_1 = \frac{A_H A_w A_\ell}{3} = \frac{0.5\,(2)(1)}{3} = 0.33 \text{ ft}^3$$

$$V_4 = \left(\frac{A_w + w}{2}\right) h_3 A_\ell = \left(\frac{2+8}{2}\right)\left(\frac{18}{12}\right)(1) = 7.5 \text{ ft}^3$$

$$\begin{cases} \text{height: } h_2 = H - h_1 = 58 - 34 = 24 \text{ in.} \\ \qquad = 2 \text{ ft} \\ \text{base width: } w = 8 \text{ ft (given)} \\ \text{base length: } B_\ell = 2h_2 = 4 \text{ ft} \end{cases}$$

$$V_3 = \frac{h_2 w B_\ell}{3} = \frac{2(8)(4)}{3} = 21.33 \text{ ft}^3$$

base length of V_2: $C_\ell = \ell - (A_\ell + B_\ell)$
$= 12 - 5 = 7$ ft

$$V_2 = \frac{h_2 C_\ell w}{3} = \frac{2(7)(8)}{3} = 37.33 \text{ ft}^3$$

$$\left.\begin{aligned} V_S &= \ell w h_1 = 12(8)\frac{34}{12} = 272 \text{ ft}^3 \\ &= 10.1 \text{ yd}^3 \end{aligned}\right\} \textit{(struck capacity)}$$

$$V_H = V_S + (V_1 + V_2 + V_3 + V_4)$$
$$= 272 + (0.33 + 37.33 + 21.33 + 7.5)$$

[See Fig. 4-56]

$$\therefore V_H = 338.5 \text{ ft}^3 = 12.5 \text{ yd}^3 \qquad \textit{(heaped capacity)}$$

$$V_H = 1.24\ V_S$$

Heaped capacity represents a 24 percent static cargo overload, based on design (struck) capacity. Naturally, a steeper heaping slope would produce a greater overload.

(d) $W_P = V_H\rho = 338.5(110) = 37{,}200$ lb

Total payload W_P is also 24 percent over the nominal design payload (15-ton) for this particular cargo density.

Equations of load height more general than Eqs. 4-65 and 4-66 can be written. One form expresses height as a variable of load length (or distance) and can be used for many cargo types. Load variation is generally based on convenient mathematical models (e.g., linear, various degrees of parabolic), some of which are more practical for vehicle loading analysis than others. An illustration of this type of formula for increasing load height only is given by

$$h = h_o x^n, \text{ in.} \tag{4-68}$$

where

h_o = load height at an initial or datum point, in.

x = load length parameter, in.$^{-n}$

n = exponent of load variation, dimensionless
($n = 0$ for uniform load distribution,
$n = 1$ for linear load distribution, and

$n \neq 0$, $n \neq 1$ for parabolic distribution).

Combination of decreasing and increasing loads and even vacant portions of cargo spaces can be accounted for by a series of such loading height equations; the design engineer can easily establish these when required. Equations of this type (Eq. 4-68) are useful in obtaining maximum height H at $x_{(max)}$ and for finding homogeneous cargo volume by integration with respect to the variable x, when height is not a variable in a direction perpendicular to x.

Assessment of load magnitudes resulting from cargo of complex shapes can be handled with the elementary $S = q = P/A$ relation (par. 4-3.2) together with some design intuition. The difficulty lies in the evaluation of the effective load-to-floor contact area—especially when load distribution pads are used. Specification of proper sizes for pads or other contacts does not in itself ensure that loads will be effectively distributed over such surfaces. Specific design analysis of the interface—pad material, thickness, relative stiffnesses—are required before load intensity calculations can be made with the elementary formula.

4-5.2 FLOOR LOADS DUE TO CARGO HANDLING

Aside from loads due to the cargo itself, vehicle floors (and other body parts) can be dynamically loaded when cargo is loaded or unloaded. Dynamic loads, in general, are treated in par. 4-5.4. Magnitude and severity of cargo handling loads depend on the type of material handling used. The human element is also present, either as the primary source (manual handling) or as the secondary source (equipment operation). Resulting loads are difficult to analyze or make allowances for in design, because most occurrences are due to accident or negligence. The best load distribution methods are useless against such practices as cargo thrown or dropped by personnel or loading equipment and impacts by self-propelled handling equipment against various parts of the structure (not primarily the floor, although the weight of a fork-lift truck working on a vehicle floor directly can cause sizable floor loads).

Little numerical guidance can be given in this area because of uncertainties in cause and severe limitations of available equations. Even for the seemingly simple case of an object dropped from a given height onto a structural member (beam), the governing strength and deflection equations are based on total conversion of the kinetic energy of the falling mass to strain energy of deflecting the beam[88].

$$\left.\begin{aligned} \delta' &= \delta\left[1 + \left(1 + \frac{2h}{\delta}\right)\right]^{1/2}, \text{ in.} \\ S' &= S\left[1 + \left(1 + \frac{2h}{\delta}\right)\right]^{1/2}, \text{ lb/in.}^2 \end{aligned}\right\} \quad (4\text{-}69)$$

where

δ' = dynamic beam deflection (due to falling mass of weight W), in.

δ = nominal beam deflection (due to static loading by a mass of weight W), in.

S' = dynamic stress due to falling mass, lb/in.^{2}

S = nominal stress due to static loading by a mass equal to falling mass, lb/in.^{2}

h = height from which mass is dropped, in.

Assumptions related to Eq. 4-69 and limitations of their usefulness are as follows:

a. Mass of the beam is negligible with respect to mass of the falling body.

b. The beam material is not stressed beyond the yield point.

c. Shape of the dynamic deflection curve is the same as that for static deflection.

d. Evaluation of static deflection δ requires additional assumption of beam and support condition.

If a further assumption is made that the respective deflections are proportional to the loading forces, Eq. 4-69 can be written in a form more useful for load considerations.

$$\frac{P}{W} = 1 + \left(1 + \frac{2h}{\delta}\right)^{1/2}, \text{ dimensionless} \quad (4\text{-}70)$$

where

P = dynamic load (equivalent static load which would stress or deflect the beam to the same magnitude as the falling load), lb

W = static load (weight of the falling load), lb

Although all the limitations concerning Eq. 4-70 will not be met in practice, it can still be used to approximate impact loads that result from severe cargo handling. The ratio of dynamic (impact) to static load P/W found for particular cases compares with load factor magnitudes associated with more common vehicle dynamic loads (par. 4-1.4 and 4-5.4).

Sample Problem

A rock of weight W = 2000 lb is dropped onto a dump body floor from a height h = 5 ft by a front-end loader. Assume the rock makes contact directly over the mid-span of an underfloor member of channel cross section, simply supported at ends of span length L = 10 ft (120 in.). (Beam effect of floor plate is negligible.) The beam member is a 5 X 1¾ standard aluminum channel, with moment of inertia $I_{x\text{-}x}$ = 7.49 in.^{4}, and modulus of elasticity E = 10X10^6 lb/in.^{2} Find the resulting ratio of dynamic to static load.

$$\delta = \frac{WL^3}{48EI} = \frac{2000(120)^3}{48(10\times10^6)(7.49)} = 0.963 \text{ in.}$$

$$\frac{P}{W} = 1 + \left[1 + \frac{2(60)}{0.963}\right]^{1/2} = 1 + 11.2$$

$$\frac{P}{W} = 12.2$$

4-5.3 HYDROSTATIC LOADS ON FLOORS

The increased exposure of military vehicles to amphibious operations makes it necessary to investigate effects of structural loads caused by such conditions. An automotive body or hull is subjected to several hydrostatic and hydrodynamic forces during the negotiation of a water barrier; these forces are treated in Section VII of this chapter. A hydrostatic load is exerted by a fluid on all submerged surfaces of a body. The load on primarily flat floor areas will be considered here. While an analysis of this type is not strictly correct for bodies underway in fluids, the dynamic effects are not expected to be significant for amphibious vehicles with slow speed performance. Many military vehicles—the lower echelons of amphibians and especially the ones limited to fording—have very nominal speed capabilities in water. Eq. 4-67 can be used to describe the pressure or uniform load intensity on the underside of a flat floor. Hydrostatic load in a given fluid is a function of immersion depth, only, on a surface oriented parallel to the fluid surface (e.g., a flat floor). Substitution of density of water in Eq. 4-67 converts it to a more useful form for amphibious considerations.

$$q = \frac{h'}{27}\ , \text{ lb/in.}^2 \tag{4-71}$$

where

h' = immersion depth of a body surface oriented parallel to the water surface (or a point in any body surface), in.

The density of seawater (three percent greater than that of fresh water, Table 4-11) was used in Eq. 4-71.

For additional information on hydrostatic floor loads see Ref. 165.

Sample Problem

Consider a tracked combat vehicle with deep-fording capability only, and a wheeled amphibious cargo truck. Typical pertinent dimensions for the respective vehicles are

a. height of turret h_{a1} = 110 in.; ground clearance h_{a2} = 18 in.; and

b. height of water line measured from bottom of tires h_{b1} = 70 in.; floor ground clearance h_{b2} = 28 in. Compare the effect of hydrostatic load on the hull and body surfaces of the two vehicles:

$$\left.\begin{aligned} q_a &= \frac{h_{a1} - h_{a2}}{27} = \frac{110 - 18}{27} = 3.41 \text{ lb/in.}^2 \\ q_b &= \frac{h_{b1} - h_{b2}}{27} = \frac{70 - 28}{27} = 1.55 \text{ lb/in.}^2 \end{aligned}\right\}$$

[Eq. 4-71]

The magnitudes of load intensity calculated here represent typical maximums, because vehicle dimensions are the governing factors. For a combat tank construction, hydrostatic loads should be of little importance; but, for standard and especially for lightweight vehicle construction, such loads are significant. The load magnitudes found in the sample problem are of the same order as those that result from typical cargo loads (see sample problems in pars. 4-2.3.2, 4-3.2, and 4-5.1). An interesting sidelight to the problem is that a net load

reduction can occur if both hydrostatic and cargo loads act on a floor at the same time, due to opposite lines of action.

Water pressure, hence hydrostatic load intensity, varies on inclined or curved floors because of the change in elevation. For these cases it is more convenient to examine a resultant pressure force acting over a defined surface and located at its center of pressure (par. 4-19). An inclined floor can be considered a type of wall and is treated in par. 4-7.4. Curved floors are not prevalent in military vehicles, but can be analyzed by considering the horizontal F_x and vertical F_y components of the pressure force separately[83].

$$F_x = \int_A q\cos\theta dA = \rho\int_A h\cos\theta dA, \text{ lb} \quad (4\text{-}72)$$

$$\begin{aligned} F_y &= \int_A q\sin\theta dA = \rho\int_A h\sin\theta dA \\ &= \rho\int_V dV = \rho V, \text{ lb} \end{aligned} \quad (4\text{-}73)$$

dA = elemental area of the floor surface, in.^{2}

dV = element of the volume formed by the floor surface, fluid surface, and four imaginary vertical planes defined by dA, in.^{3}

V = total volume formed by the elements dV, in.^{3}

ρ = fluid density, lb/in.^{3}

θ = angle between the fluid surface and the normal to dA, measured from the former, deg

The physical interpretation of Eq. 4-72 is that the elemental areas of the curved surface are projected horizontally onto any flat vertical surface, and summed to give one horizontal pressure force component acting over the total area; a very similar procedure is used for working with walls (par. 4-7.4) and formal integration is seldom needed even for most curved surfaces. Physically, Eq. 4-73 projects elemental areas vertically onto a specific horizontal plane—the fluid surface; thus, elemental volumes are formed by the products of elemental areas of the fluid surface and the elemental fluid heights. Summation of all the elemental volumes or the "projected volume" of the curved surface is proportional to the other (vertical) pressure force component.

4-5.4 DYNAMIC LOADS ON FLOORS

The body or hull of a military vehicle experiences many complex applied loads that cannot be analyzed by normal static-oriented procedures—indeed, some of the dynamic effects cannot be adequately treated by any known technique. Many, but not all, dynamic loads are due to the particular motions of vehicles in the military environment. The sources of specific dynamic loads are many and a number of them were treated in par. 4-1.4 relative to the vehicle structures as a whole. Of these dynamic loads, terrain-imposed and cargo-imposed loads are especially significant to floors of bodies and hulls.

4-5.4.1 Terrain-imposed Loads

Terrain as related to the military environment has been described in par. 1-2. In vehicles with an integral frame and body construction, the dynamic forces set up at ground-contacting elements pass through the suspension system directly to the floor structure. In vehicles with separate frame and body construction, these dynamic forces also pass through the frame. A certain amount of attenuation takes place between the point of application of the shock and the point under consideration. In general, components mounted within a vehicle will experience shocks of lower intensity than will the body or hull which will, in turn, experience less shock than will the axles or wheels, as a result of the attenuation produced by the suspension system. This is not always the case, however, and depends upon the location of the component in the vehicle (see par. 4-1.4.1 and Fig. 4-3).

The problem becomes extremely complex since the initiating ground forces are not yet calculable analytically. Experimental methods have provided useful information about terrain loads, but present techniques are not completely adequate for the vehicles under discussion here[84]. More work along experimental lines is needed and also analytical followup to correlate, amend, and otherwise tie together generated data.

One semi-empirical method[85] for estimating terrain-imposed loads is available and is more completely described in par. 4-1.2.1 In that method, a "basic load factor" is determined

using a nomograph that takes into account important vehicle parameters; namely, vehicle type, mission, and gross weight. Load factors so obtained represent maximum accelerations (in g's) on bodies and hulls of vehicles represented in the nomograph. For design purposes, dynamic loads can be approximated by use of the load factor as a multiplier. Maximum acceleration will vary with position along a body and values as read from the nomograph need to be modified for certain vehicle types (see par. 4-1.2.1 for details).

Use of load factors or acceleration-type multipliers to calculate dynamic loads is based on Newton's second law of motion

$$F = ma = \left(\frac{W}{g}\right)a = \frac{W(ng)}{g} = nW, \text{ lb} \qquad (4\text{-}74)$$

where

m = mass of the body under acceleration, lb-sec^2/ft

a = acceleration of the body, ft/sec^2

W = weight of the body under acceleration, lb

g = acceleration due to gravity, ft/sec^2

n = load factor or multiplier used to express a specific acceleration in terms of g, dimensionless

The procedure of using the g-multiple of the peak accelerations as a multiplier in determining the magnitude of the shock forces is a popular, but technically unsound, method. The difficulty lies in the correct assessment of the weight being accelerated since components of different weights subjected to the same shock or acceleration will experience dynamic forces proportional to their weights. Use of incorrect "effective accelerated weight" results in vehicles capable of safely withstanding sustained loads many times greater than those normally experienced by the vehicle. This practice tends to produce overdesigned vehicles with the attendant excess weight and excess cost.

A second, undesirable aspect of overdesigning is the increased stiffness of the overdesigned members. This increased stiffness results in an increase in the natural frequency of the structure. Since the acceleration inherent in a vibrating mass is proportional to the square of its frequency, increasing the stiffness of a structural member can decrease its resistance to continuous vibration.

Natural frequencies found in the body structure of some military transport vehicles are listed in Table 4-12. It is important to note that the data represent rough road operations at speeds above which vehicle damage and operator hazard were probable[86].

A sizable problem also associated with terrain-imposed loads is the racking or twisting in a body or hull. Advances in suspension design have, undoubtedly, reduced such effects, but, under true cross-country operations, even military vehicles with the most modern suspension systems undergo severe roll motions that cause racking. Again, the inherently heavier, closed construction of combat tanks, personnel carriers, and other armored vehicles is sufficient to counteract this dynamic load source. Even the lighter, but completely closed, or tubular construction of tank-type bodies exhibits resistance to torsion. Lightweight vehicles with closed transverse sections (vans) and many open section transport vehicles are weak with respect to torsional loading, however. One promising solution to body and hull torsional problems has been presented and is a byproduct of continuing efforts to improve vehicle mobility; this developmental item is a tractor-carrier combination vehicle connected by an articulated (double universal joint) coupling. An example of a military vehicle built with this concept is the XM561, 6 X 6, 1¼-ton Cargo Truck. This experimental vehicle features an advanced articulation system whereby the carrier is allowed to roll in addition to pitch through significant angles with respect to the tractor—with the creation of only negligible torsional body stresses (Fig. 4-57). The roll capability is, of course, the important point in the limiting of racking. The advantages of this vehicle type when fully developed are obvious from Fig. 4-57 both to structural and mobility viewpoints.

4-5.4.2 Cargo-imposed Loads

The various dynamic loads which act on vehicle structures also produce reactions on their contents. Although both personnel and material cargo can suffer adverse effects from some of the more violent loads and motions, only the latter cargo type is of interest here because of its

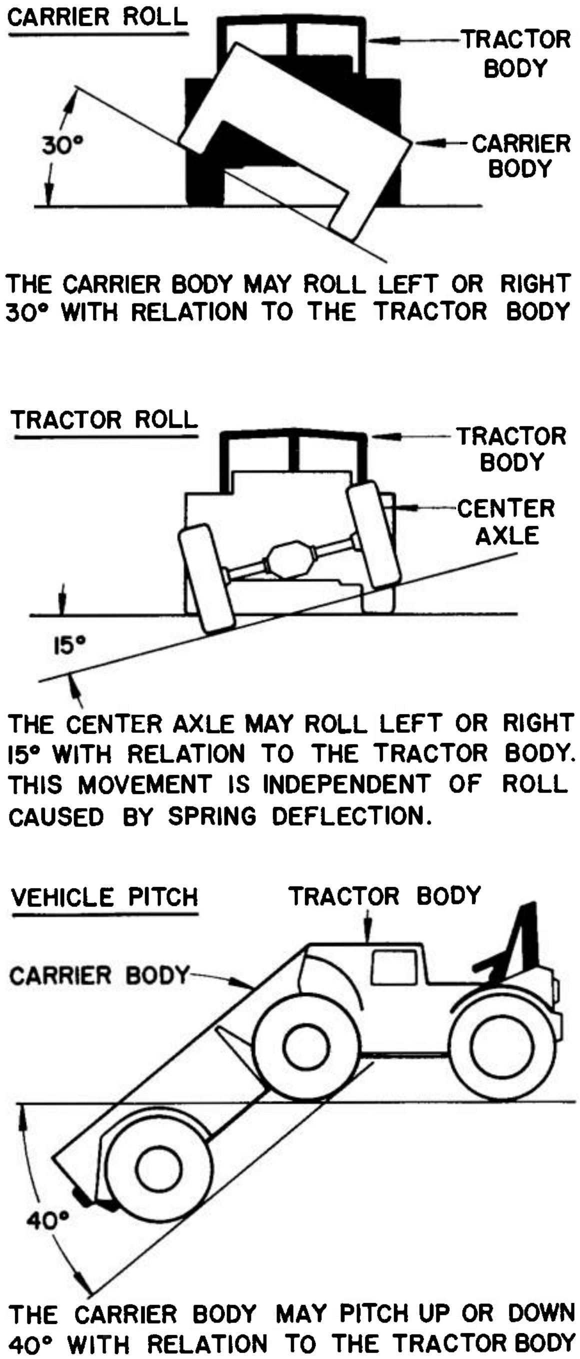


Figure 4-57. XM561 Articulation Attitudes[87]

damage potential to structures. It is not hard to visualize that unfastened cargo will undergo violent jostling, bouncing, and other complex motions during typical military vehicle operation; these cargo motions induce severe shock loading on body and hull parts. With adequate cargo tiedown methods (par. 3-41), independent cargo motions are eliminated but adverse accelerations remain to cause dynamic loads. For lack of better methods, the use of empirical load factors or measured accelerations in conjunction with Eq. 4-74 is offered for determination of dynamic cargo loads. In this situation the weight under acceleration is known, and an analysis to determine the "effective accelerated weight", as for road loads, is not needed. To make Eq. 4-74 fully usable, specific load factors or cargo load factors are needed. One example of this type of data is given in Table 4-13 which lists cargo acceleration in a 2½-ton truck and M104 Trailer combination for various terrain. Road load factors are given in Fig. 4-2 and are discussed more fully in par. 4-1.4.1.

TABLE 4-12 PREDOMINANT FREQUENCIES MEASURED IN CARGO SPACES OF VARIOUS MILITARY TRANSPORT VEHICLES[88-91]

Type of Vehicle	*Direction of Acceleration*	*Predominant Frequencies, cps*		
		Body (Sprung Mass)	*Vibration on Tires*	*Resonant Vibration of Structural part of the Body*
Truck (2½-ton)	Vertical	2-4	8-13	70-180
	Longitudinal	—	10-20	70-100
	Lateral	2	10-20	100-200
Truck (3/4-ton)	Vertical	2-3	5-10	60-110
	Longitudinal	—	—	70-100
	Lateral	—	—	60-70
Trailer (1-ton)	Vertical	3-5	8-10	50-100
	Longitudinal	—	—	50-100
	Lateral	2	—	50-120
M14 Trailer	Vertical	1-4	7-10	50-70
	Longitudinal	3-4	8-10	200
	Lateral	2-4	—	—
M1 Trailer (2-ton)	Vertical	2.5-5	7.75-10.5	100-150

TABLE 4-13 CARGO ACCELERATION IN 2-½-TON TRUCK M104 TRAILER COMBINATION[89-91]

Operation Over	*Maximum Acceleration, g*			
	Longitudinal	*Lateral*	*Vertical*	*Vector Total*
Sandy Beach	2.5	1.0	4.5	5.3
Ungraded Road (30 mph)	0.5	1.0	1.5	1.9
Graded Road (30 mph)	1.0	0.25	1.0	1.4

SECTION III—WALLS—CONSTRUCTION AND DESIGN

4-6 GENERAL CONSIDERATIONS

The extent of walls in military land vehicle structure is a great variable because of close relation to varied body and hull types (Chapter 1). The range extends from platform bodies with no walls at all, to massive walls of tanks and other combat vehicles, and to walls of very sizable surface area in van bodies. Wall proportions are governed by need or function in a given vehicle or mission. General size limits are dictated for wall structures. These limits must consider such factors as overall vehicle height, center-of-gravity height, introduction of problems of lateral strength and stiffness, and overall weight and cost. From a tactical viewpoint, increase of overall height (and length) makes a vehicle more vulnerable to enemy observation and fire, and increases the cargo space required to transport it. Increase of center-of-gravity height makes the vehicle less stable, particularly on side slopes and when cornering. Tactical form and size considerations may be more severe than physical limits (par. 2-1.).

Aerodynamic losses in vehicles depend on speed and form. This area has been of little concern to the military land vehicle designer, and by and large rightly so, because the vehicles in his responsibility traditionally operate at relatively low, sustained speeds. The speed factor can be expected to change in the future under the influence of continued development of conventional and unconventional technologies. Recent advances in suspension systems that have markedly increased cross-country performance and work with ground-effects vehicles[92] are but two indications of higher future speeds. At highway operation, military vehicles have long been in the region of aerodynamically significant speeds and have paid penalties in power requirements. Poor aerodynamic form, the other influencing factor, has always been present and is the least favorable in van bodies. As an example of the power losses incurred due to wind resistance, a 55,000-lb GVW truck-van semitrailer unit traveling at 60 mph was found to require 200 hp at the wheels—of which 100 hp was necessary to overcome wind effects[72]. At legal highway speed limits, aerodynamic and mechanical power requirements become approximately equal in importance. Wall structures are not necessarily the worst offenders; roofs, frontal areas, body protrusion, and especially exposed underbodies contribute substantially to wind resistance. Some recent military vehicles have received a sort of underbelly "streamlining," but this construction change was made for improved amphibious operations and better obstacle traverse. Aerodynamics will play an increased role in structural design of military vehicles, including walls, when additional power requirements of future speeds make this extra effort necessary.

4-6.1 TYPES OF WALLS

Type of wall construction, as floor construction, is determined largely by the body or hull type of a particular vehicle. In a given vehicle, wall construction will be quite similar to floor construction (par. 4-4.1) but material gages and thicknesses will be thinner in many instances. Exceptions to this generalization are found in semimonocoque structures, dump bodies, and hulls of combat vehicles, i.e., in situations where wall loading and floor loading are equally important. Specific walls can be composed of several elements, but most may be classed with respect to the form of protective cover they exhibit. Distinct types of wall covering are (a) castings and plates (material thickness greater than 1/4 in.), (b) flat and formed sheets (thickness greater than 1/4 in.), (c) extruded planks, (d) double-wall components of sheet and extruded forms, and (e) sandwich panels. The covering surfaces are normally supplemented with various stiffeners such as columns, posts, and rails. The extent of additional reinforcement needed depends upon how well the wall covering is also used as a working or load-carrying member. Inherent strength of thin sheet metal construction against certain loads is often overlooked (par. 4-2.3.3).

Castings and plates represent the heaviest construction type for walls and other portions of structures. This construction is limited to combat vehicles where the strength and thickness of components must withstand blast

and ballistic impact loads, in addition to other more common dynamic vehicle loads. As a result, more excess structural weight must be tolerated in this vehicle class. Even in this area, traditional construction with its great weight penalties is undergoing reappraisal; the plate-weldment type of hull rather than the still heavier cast hull is finding acceptance. Figs. 4-48 and 4-49 show wall construction in typical combat vehicles, using castings and plates, respectively. Particularly here, there is very little to distinguish between floor and wall construction. For more information on this subject see Ref. 17.

Flat and formed sheets make possible considerable versatility in construction and this material finds application in many body types. Outside of combat vehicles, there are few wall structures that receive sizable direct (normal) loads. One notable exception is the dump body, where heavy wall loads are developed as a result of loose bulk cargo. Other wall structures, e.g., on military cargo and van bodies, play more of a protective and sheltering function; in these cases, sheet gage is based on other than strength considerations. Even in semimonocoque construction where loads are transmitted from the floor to walls, thin sheet material is made to work by use of its favorable tensile and shearing strength (par. 4-2.3.3). The convenience of sheet metal gage selection to suit various load conditions or other considerations, e.g., puncture resistance, is an obvious advantage.

A typical wall will consist of large sheet sections riveted or welded to a framing of structural shapes. More generalization is not possible because a large number of variables exists from one body type to the next. Some characteristics to be found in the three major types with walls (military cargo, dump, and van) will be mentioned. Extent of support framing varies from the very elaborate in a van body (Fig. 4-58) to much fewer, more robust, mainly vertical framing in cargo and dump bodies. Fig. 4-58 also gives terminology pertinent to automotive wall and roof construction. Frame components will be standard structural shapes, but top-hat sections and rectangular cross section tubing are commonly used. The importance of shape selection in design is considered in par. 4-2.2. Cargo and dump bodies are externally framed with an inner sheet skin to present little obstruction to cargo loading and unloading; van bodies are internally framed which makes possible a simple and economical overlay of either flat or corrugated sheet wall material as shown by Figs. 4-59 and 4-60, respectively. Sheet material size and orientation are of little concern for cargo and dump bodies since the low wall height and even the length can be spanned by a single piece. If vertical wall joints are necessary, they will have short lengths and can be easily accomplished by standard joining methods. In contrast, van body walls have substantial height often combined with great length. This requires long joints in the wall sheets in one of two directions. Joint location is determined by the orientation of individual sheets with respect to the whole wall and to the framing; both horizontal and vertical sheet layouts are common. Aerodynamic effects enter the picture when corrugated sheet walls are given similar directional orientation. Obviously, the vertical orientation of corrugations shown in Fig. 4-60 creates more air turbulence for a given flow velocity than a horizontal orientation which is parallel to the flow. Par. 4-6 gives a brief discussion of aerodynamics as related to military land vehicles and particularly the van-type body.

Comparison of flat to corrugated sheet gives advantage to the latter on the basis of stiffness per weight, but it is less economical and less convenient for attachment to stiffeners. Embossing techniques are used with flat sheet to increase stiffness with ribs or other formed contours; this offsets the advantage of corrugated sheet but retains the properties of simple edge attachments.

Extruded planks are available mainly in aluminum in many thin-wall cross sections, but most section shapes and the methods for assembling such long, yet narrow members are more suitable for floor rather than wall construction (par. 4-41). Some corrugated shapes are suitable for low height walls in other than van bodies, however. Since only a few planks are necessary to obtain wall height, fabrication including framing does not mean excessive riveting or welding. Plank construction enjoys the advantage of one-piece lengthwise parts between wide dimensional limits; producibility considerations are eased, especially if different model sizes or changes are contemplated, because the planks can be cut to desired length locally. A better stiffness/weight

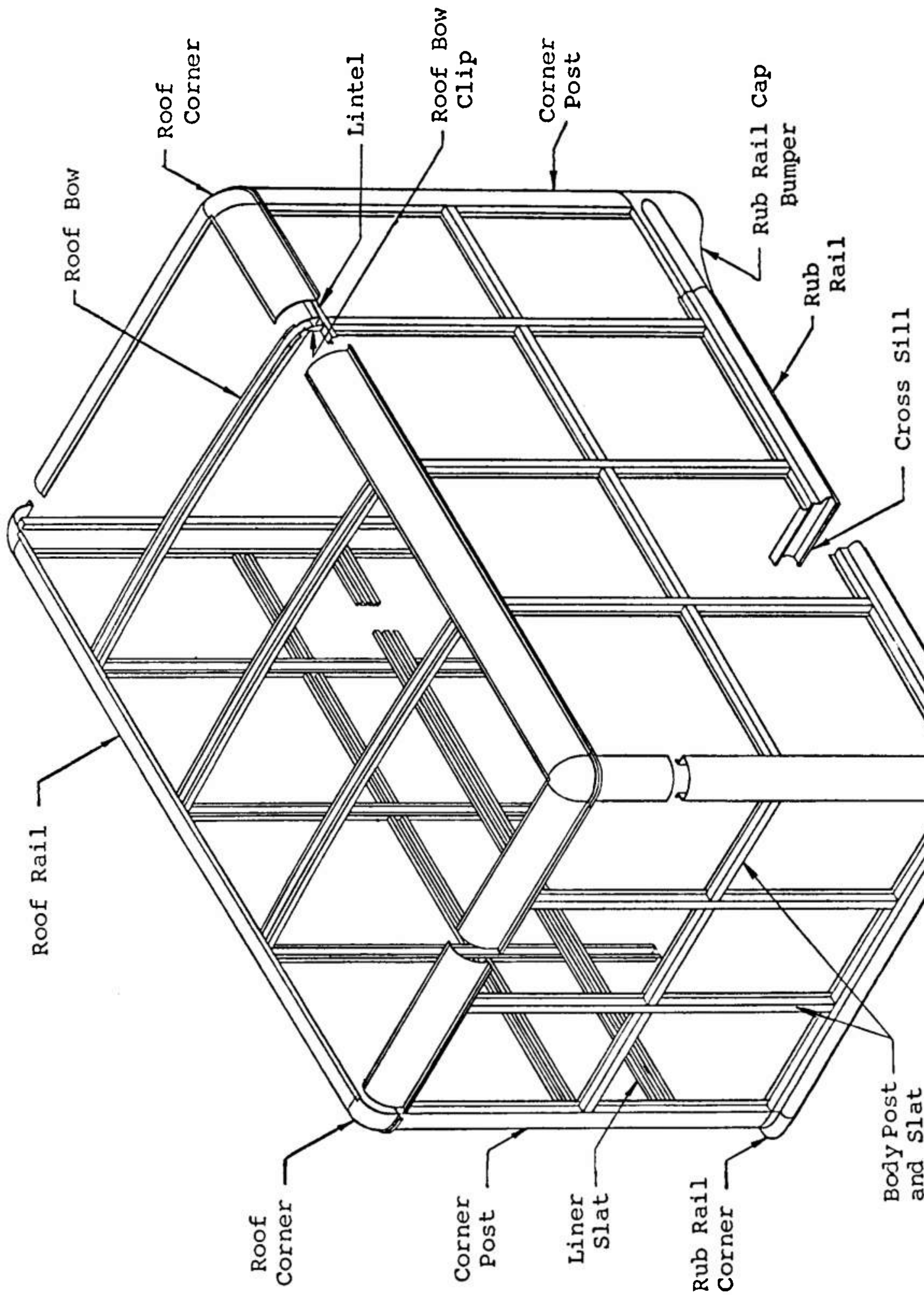
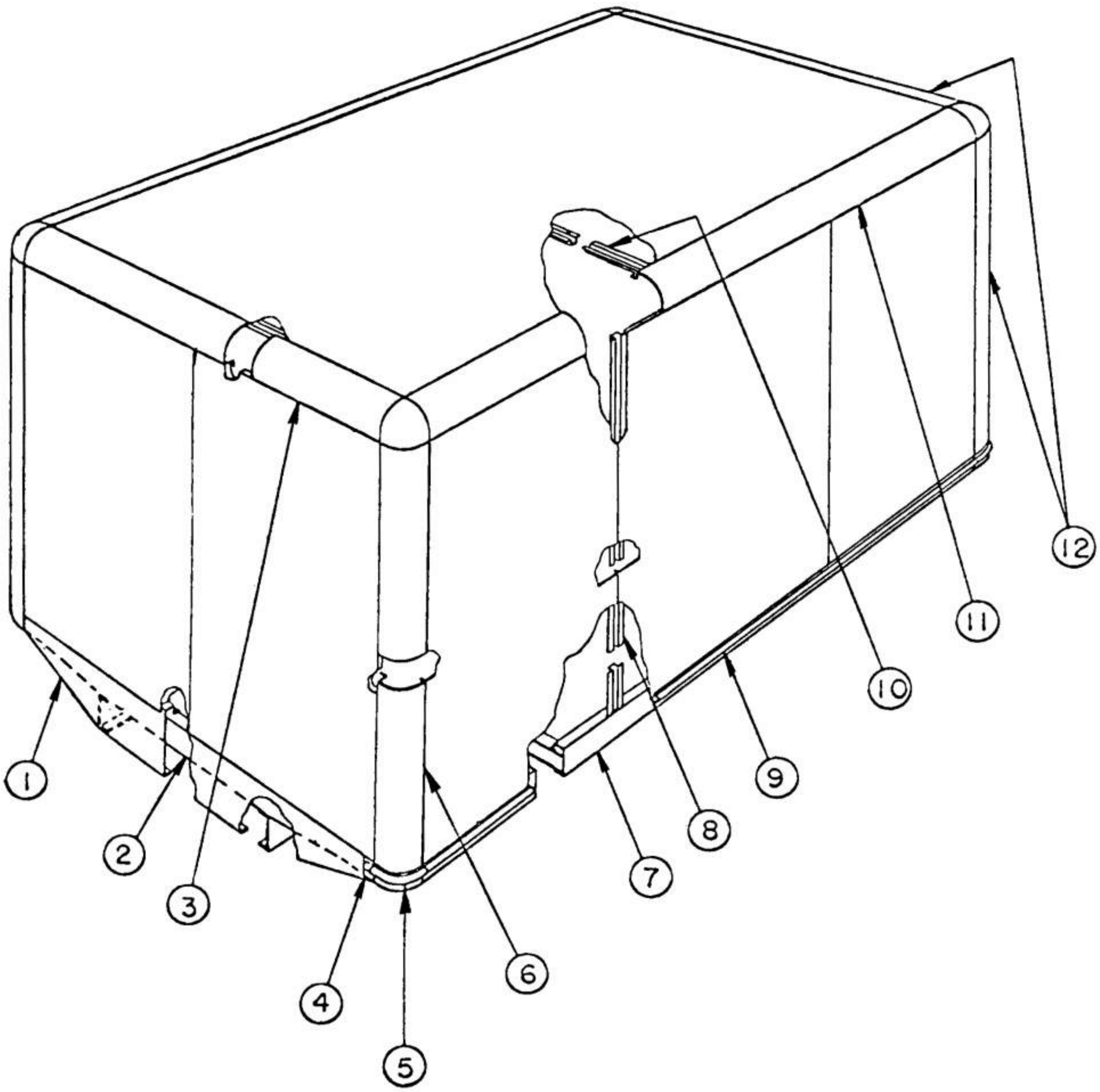


Figure 4-58. Typical Wall and Roof Construction, Van Body[96]



1	Bumper	5	Rub Rail Corner	9	Rub Rail
2	Tapered Cross Sill	6	Corner Post	10	Roof Bow
3	Roof Rail	7	Rub Rail	11	Roof Rail
4	Rub Rail Cap	8	Body Post	12	Corner Post

Figure 4-59. Flat Sheet Wall Construction[96]

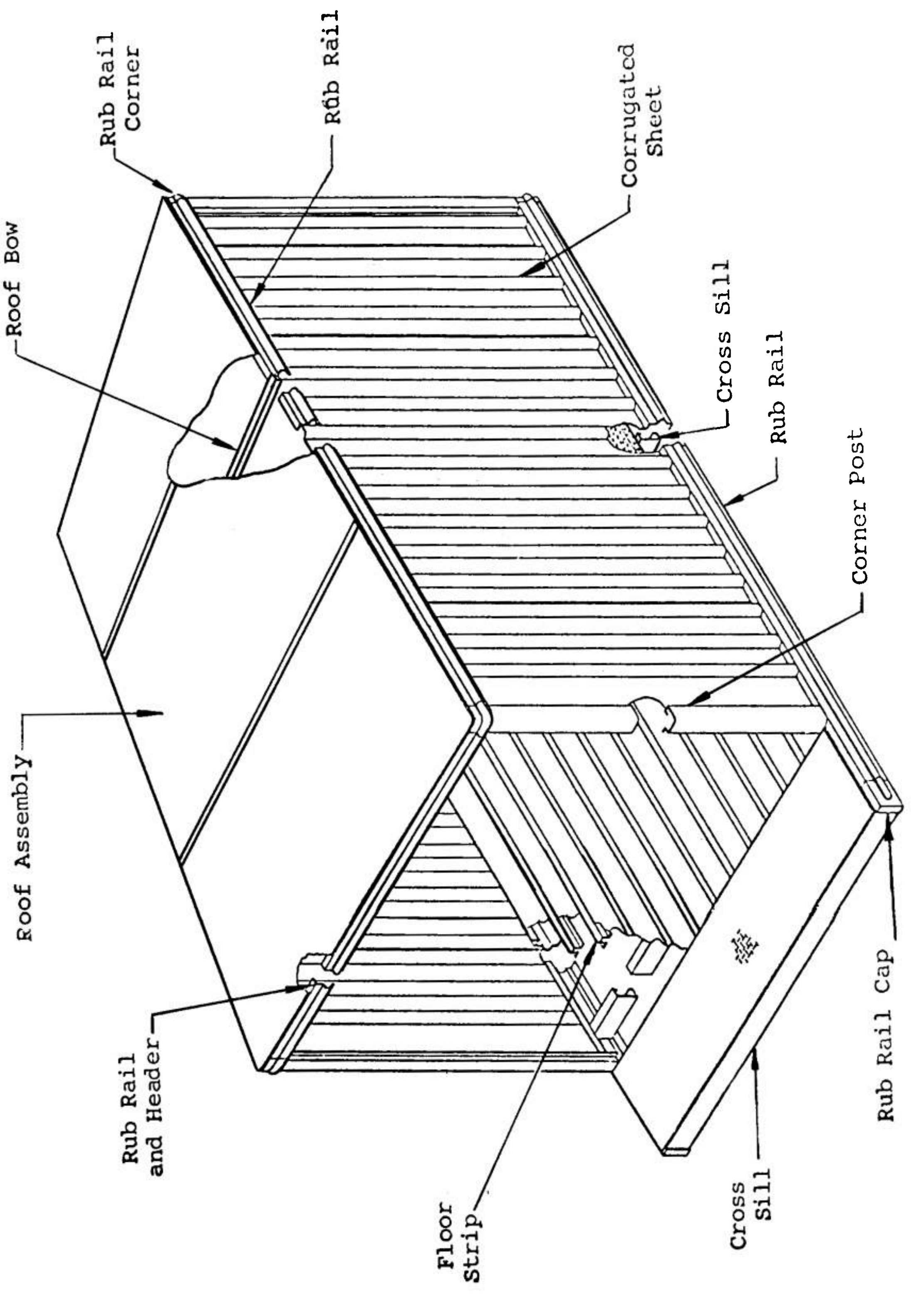


Figure 4-60. Van Body, Corrugated Sheet Wall Construction[96]

ratio is obtained than in sheet construction.

Double-wall components make possible significantly higher strength and stiffness per weight than other forms of wall construction. This is accomplished by forming thin-walled, closed sections with sizable moments of inertia within a wall panel—by sheet-type fabrications or by use of special, hollow extruded planks. Panels built up from sheets can range in configuration from the simple to the most complex. An inner and outer sheet skin, capped by channel-like members around all sheet edges and internally braced by similar members, is a simple configuration applicalbe to cargo body sides. This construction is illustrated for a cargo body drop-side in Fig. 4-61(B). More elaborate built-up panels are used in bodies of van and panel trucks, busses, and passenger automobiles. Especially in the latter examples, the sheet material is shaped by sophisticated forming methods prior to panel assembly. Enhanced styling and strength features that result are offset by high manufacturing costs. Styling is an unnecessary consideration for military vehicles, and reasonable wall strength increase can be obtained using flat sheets as a double skin. In typical installations, the inner skin is made of somewhat heavier gage to guard against cargo loads and accidental internal impacts. The space between double skins is convenient for placement of insulating material to provide environmental insulation for personnel or cargo.

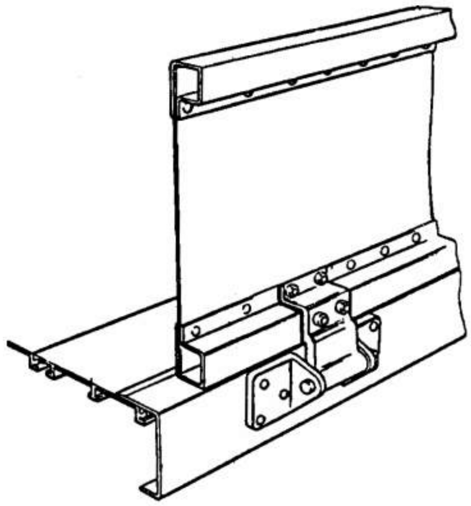
Extruded planks of closed section or hollow planks can be basic building blocks for low height walls. One example of this structural component in aluminum[93,97] is shown in Fig. 4-62(A) along with important shape selection parameters (par. 4-2.2) and weight data; contours of the plank provide for tongue and groove joints between mating parts for efficient fabrication. Fig. 4-62(B) shows a typical double-wall plank assembly for a cargo body side. As in other types of low wall construction, the wall top is capped by an angle or channel section; a top-hat section is used as the outer, vertical bracing.

Sandwich panels with low-density cores are very efficient structural components. Although these materials are still undergoing development for military land vehicles, their application to wall construction has been shown. (See par. 4-2.3.2 for sandwich panel design.)

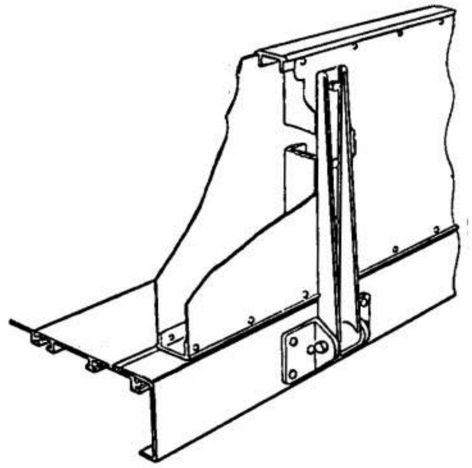
Drop-sides and tailgates, which are essentially hinged walls, can be found on military cargo bodies; tailgates are also found on dump bodies (pars. 1-5.1 and 1-5.3). The proportion of such hinged sections to the entire wall structure can be very large on the new concept cargo vehicles. Hinged sections must be sturdy enough to take loads when acting as walls and yet be light for hinging and handling. Generally, the construction described previously in connection with fixed walls applies to hinged walls; but the more lightweight methods are obviously preferred. Fig. 4-61 shows typcial drop-side and tailgate fabrication details. The efficient use of aluminum extrusions as top and bottom framing sections should be noted in the figure, Single-skin construction (Fig. 4-61(A)) will require additional vertical supports that are not shown. Double-skin construction (Fig. 4-61(B)) needs fewer vertical supports which are located between the skins. Hollow plank construction (Fig. 4-61(C)) brings with it both structural and production efficiency that can be utilized in hinged as well as fixed walls. Framing is required to protect and close exposed ends of structures when using any of the three fabrications.

Inner wall liners are commonly used on van bodies. Linings serve several functions. They protect the walls from cargo damage and accidental impacts caused by portable materials-handling machinery. Interior wall stiffeners, similar to floor skid strips, are sometimes employed, in addition, to protect against internal wall impacts. In addition, linings are used to eliminate obstructions from the inner framing typical to vans, when a smooth interior is required; to provide environmental insulation; and to serve as a convenient attachment surface for wall-mounted equipment. Metallic and nonmetallic sheets are the most common lining materials, but a plank or slat-type liner is also used (Fig. 4-58).

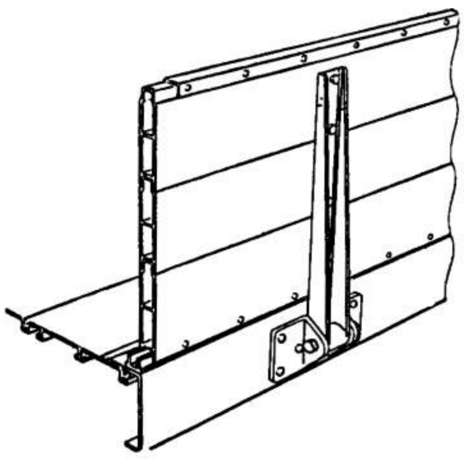
Extruded aluminum sections are used extensively as strength members in modern land vehicle structures, because this manufacturing process makes a large variety of useful shapes readily available. The extrusion method is not yet commercially available for steel, but production by roll forming is highly advanced in this material. Roll forming can produce many desirable shapes, but fewer than are possible with extrusions. Many publications of metal



(A) Single-Skin Construction

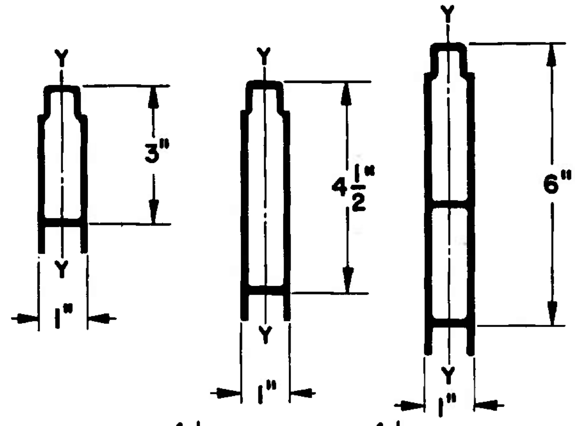


(B) Double-Skin Construction



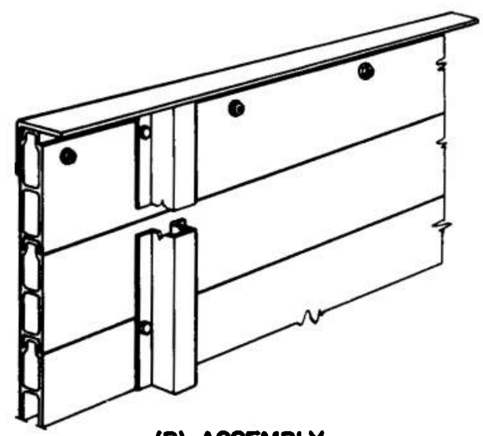
(C) Hollow Plank Construction

Figure 4-61. Wall Construction—Drop-sides and and Tailgates[97]



I_{yy} 0.131 IN.^{4}	I_{yy} 0.206 IN.^{4}	I_{yy} 0.274 IN.^{4}
Z_{yy} 0.262 IN.^{3}	Z_{yy} 0.412 IN.^{3}	Z_{yy} 0.548 IN.^{3}
GENERAL THICKNESS 0.1"	GENERAL THICKNESS 0.1"	GENERAL THICKNESS 0.1"
WT/FT 0.9 LB	WT/FT 1.4 LB	WT/FT 1.6 LB

(A) BASIC COMPONENTS



(B) ASSEMBLY

Figure 4-62. Plank, Double-wall Construction[73,97]

manufacturers and of automotive structural part producers are specifically directed toward the vehicle structures designer; one source in the latter category is Ref. 94. The various strength members in wall structures are found in the internal, or external, framing characteristics of certain body types, as mentioned earlier. External framing becomes very heavy on bodies—e.g., dump bodies—that are prone to severe loading and unloading procedures, impacts, or minor collisions. Certain of the longitudinal frame members have the function to protect body side panels and tires (duals) from a sliding type impact with walls, terrain obstacles, and other vehicles. These components are called rub rails and extend out laterally

beyond all body and running gear parts. Another use of outstanding external strength members is the prevention of detrimental structural damage as a result of vehicle overturn or up-ending in tactical operations. Figs. 1-6 and 1-7 illustrate rub rails on typical vehicles. Even internally framed walls are given the extra protection of rub rails as shown in Figs. 4-58 to 4-60. Lifting eyes are often located on rub rails because they provide favorable position and strength.

Useful body construction details are found in Military Specifications; the design engineer should refer to subject lists of specifications on particular vehicles. For guidance, typical minimum wall thicknesses and compositions are presented in Table 4-14, as obtained from randomly selected Military Specifications.

4-6.2 WATERTIGHTNESS OF WALLS

As mentioned in connection with floor watertightness considerations (par. 4-4.3), the largest problems in this area occur at floor-to-wall joints and at the extensive wall openings found in certain automotive structures. The very difficult sealing areas exist in wheeled transport vehicles as a result of construction methods and body features. Part of the problem arises from the more open, inherently less amphibious quality of a standard wheeled vehicle. Fording, the negotiation of a relatively minor water barrier, means potential hazard to cargo within nonamphibious bodies of wheeled vehicles. Comparable tracked vehicles will have substantially deeper fording capability even without fording kits (par. 3-36.2). With fording kits, the latter vehicles still enjoy considerable fording depth advantage over the former because of construction type. Fording kits are used to seal body and hull joints and access openings to prevent or reduce entrance of water. Again, because of construction differences, tracked vehicle fording kits are simpler than those for wheeled types.

Watertightness requirements for fording are not severe since exposure is temporary; the effectiveness demanded of fording kits is specified only for a 15-minute duration in Ref. 76. In trailers, however, even this seemingly nominal requirement presents problems, along with weather sealing (par. 3-7.1) against rain, snow, and dust. Floating capability poses more severe requirements for land vehicles; the sealing demands are more stringent and not set by exposure time limits. Fording kits combined with generally cumbersome flotation devices (par. 3-36.3) are used to limit water seepage. In some combat and tactical vehicles, a limited and specified water influx rate is allowable. This seepage is discharged from the vehicle by means of bilge pumps[96].

All-welded hull-like construction has been the principal means of assuring complete watertightness in both tracked and wheeled amphibious vehicles. This construction method has done a great deal to move wheeled vehicles closer to tracked types in amphibious capability. Another part of the problem—body features adverse to structural integrity—continues to plague even certain types of the new, developmental wheeled vehicles. Certainly one of the most severe examples of this is found in the new XM656 (5-ton) and XM410 (2½-ton), 8 X 8 Cargo Truck, which features nearly full-length drop sides in addition to a full-width tailgate. These provisions are advantageous to rapid cargo handling, but at the same time create a tremendous sealing problem during amphibious operations, as clearly illustrated in Fig. 4-63 for the XM656 Cargo Truck. The method used to keep out water is a system of inflatable seals provided around the drop-side and tailgate (also cab) openings; the seals are inflated by compressed air before the vehicle enters the water[97]. A system of this type is complex by its nature and is a current design problem area. Leaks at the various cargo body

TABLE 4-14 TYPICAL VEHICLE WALL THICKNESSES

Body Type	Vehicle Size	Construction	Minimum Thickness, in.	Gage*	Reference
Dump	17-ton GVW 4 X 4 truck	Sides Front (head) panel Sheet Steel Tailgate	0.1644	8	MIL–T–26150D (USAF)
	12-ton GVW high lift 4 X 2 commercial truck				MIL–T–46797 (MO)
Van	2½-ton 6 X 6 Series M44 and M133 Trucks	Wall panel (stretcher-leveled sheet steel)	0.0359	20	MIL–T–712A
	5-ton, 6 X 6 Series M39 Truck	Inner liner (plywood) Total Wall	1/4 ~9/32	—	MIL–T–740A

*Manufactuers' standard gage for steel

openings have been found in vehicle developmental tests as well as difficulties with a bilge pump system that is provided to discharge any water seepage past the seals. An additional problem area for the military cargo body during amphibious operations is its low wall height. It ships water rather easily from wave action, splash, and when leaving a water barrier due to low freeboard. For example, the XM656 Cargo Truck (Fig. 4-63) has only a 9-in. freeboard when swimming fully loaded.

4-7 WALL LOADS

4-7.1 CARGO-IMPOSED LOADS ON WALLS

Cargo loads on walls occur primarily by direct contact but also from indirect causes. Intentional contact between wall structures and cargo occurs in relatively few instances in military land vehicles. Significant wall loads are, therefore, limited to certain vehicles. This should not be taken to mean that dynamic loads between nonrestrained or shifting cargo and walls are equally scarce; but use of adequate cargo tiedown practices (par. 3-41) whenever practicable should effectively reduce such wall loads. Dynamic cargo loads are specifically discussed in the paragraph which follows. Cargo loads on walls, as treated here, deal with continuously applied loads similar to floor cargo loads. It should be noted that dynamic wall loads are generated from this load source when a vehicle is in motion; since this type of wall loading is analogous to dynamic floor loading it can be treated as outlined in par. 4-5.4.

The most severe, continuously applied wall loads are encountered on dump bodies. It is obvious from Fig. 4-56 that significant wall loads are developed due to the nature of load distribution in loose bulk cargo. If a vehicle of this type is used for its intended mission, many of the materials it transports will have approximately uniform density (Table 4-11). These materials obey the laws of hydrostatics that pertain to pressure forces and load distribution. Therefore, a reasonably uniform pressure is exerted in all directions at a given depth of material. This means a wall is subjected to linearly increasing pressure or load intensity which is a function of depth of loading, as shown in Fig. 4-64. Eq. 4-67 is applicable in this situation also, except maximum wall load intensity is given by

$$q_{(max)} = h_1 \rho, \text{ lb/in.}^2 \tag{4-75}$$

where

h_1 = inside height of sides of cargo box, in.

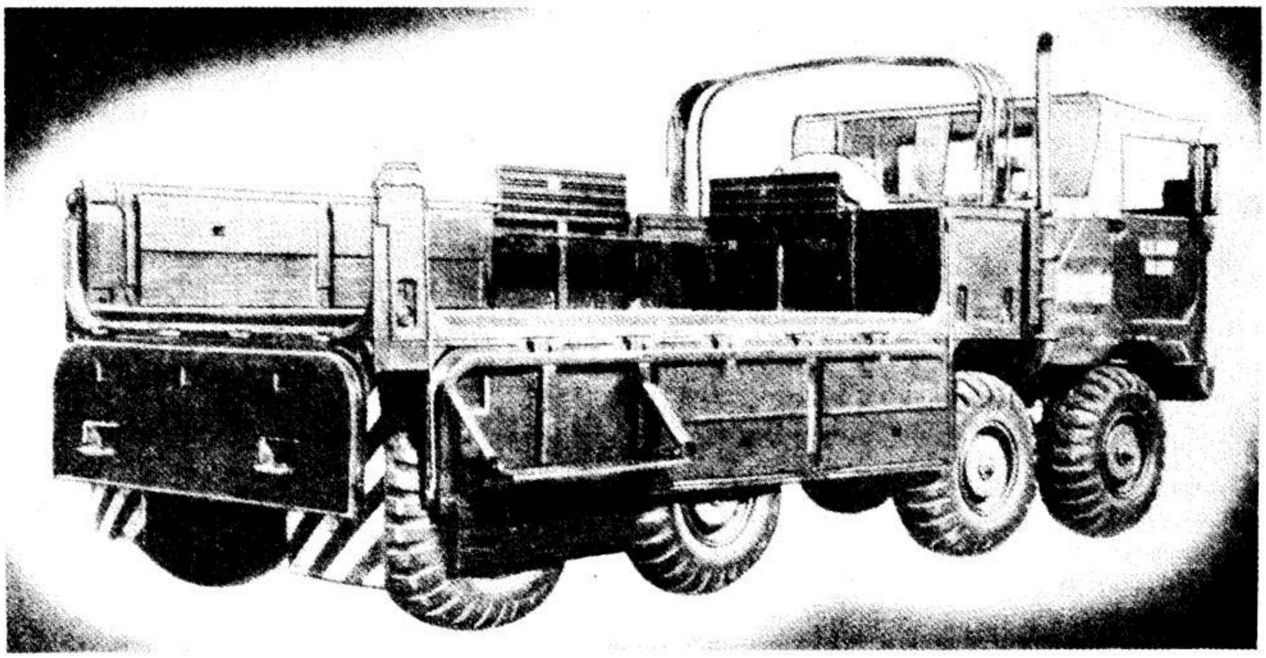


Figure 4-63. Hinged Wall Structures, Drop-sides and Tailgate, XM656, 5-ton 8 X 8 Cargo Truck

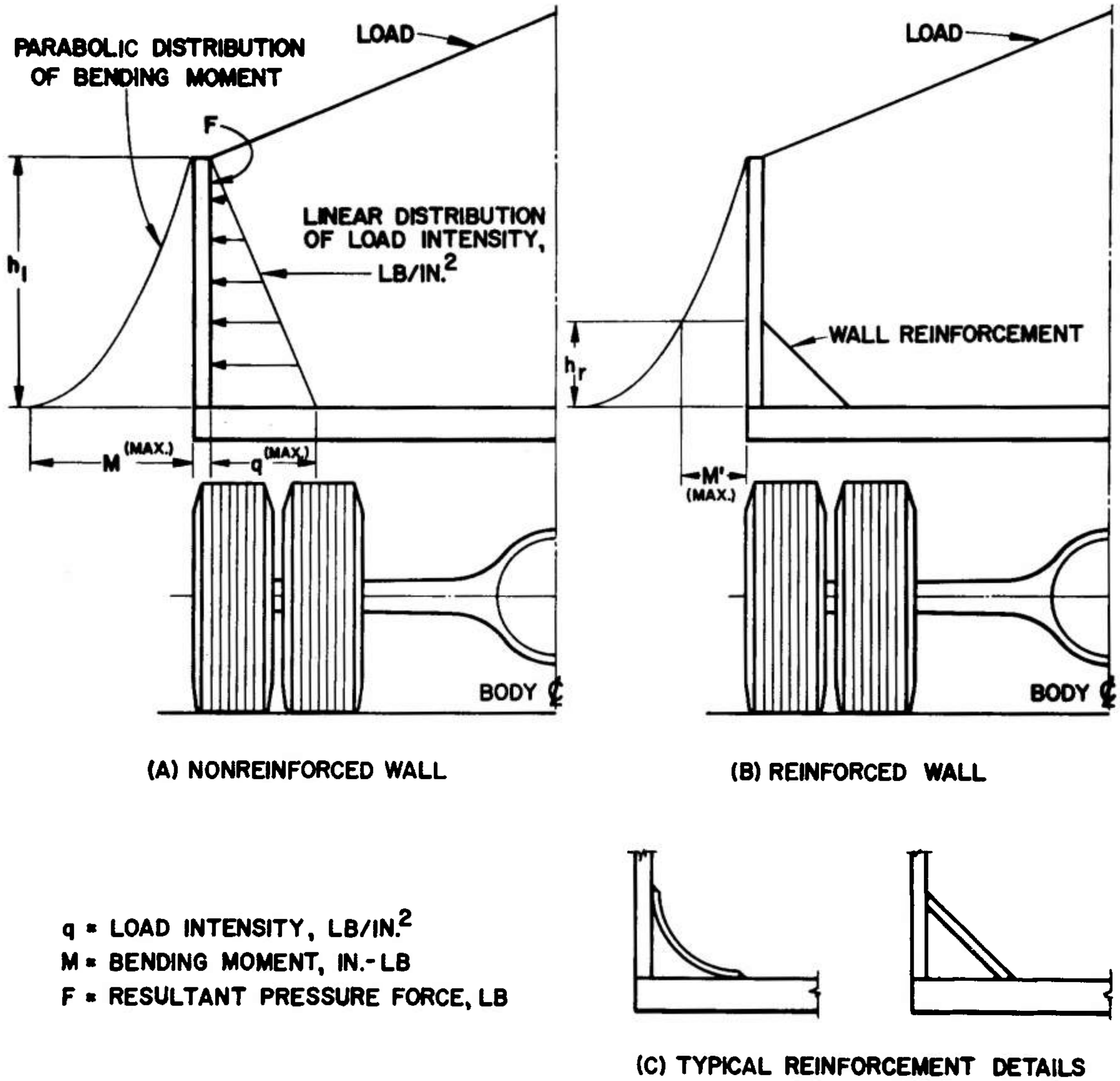


Figure 4-64. Wall Loads, Dump Body

ρ = density of material, lb/in.^{3}

It is convenient for some computations to consider a resultant pressure force acting on the wall; this force is a function of the pressure prism shown in Fig. 4-64(A).

$$F = \left[\frac{q_{(max)}}{2}\right] h_1 \ell = \frac{\rho h_1^2 \ell}{2}, \text{ lb} \qquad (4\text{-}76)$$

where

ℓ = length of the wall, in.

The pressure force will have a line of action perpendicular to the wall and will pass through the centroid of the pressure prism, or at a height $h_1/3$ measured from the cargo floor. An applied load with linear distribution produces a parabolic bending moment distribution in the wall. Maximum bending moment $M_{(max)}$ and, hence, maximum bending stress will occur at the bottom of the wall. For the nonreinforced case (Fig. 4-64(A)), the equation

$$M_{(max)} = \frac{q_{(max)} h_1^2}{6} = \frac{\rho h_1^3}{6}, \text{ in-lb/in. of wall length} \qquad (4\text{-}77)$$

is useful for approximate calculations only, since it treats the wall panel as merely a cantilever beam; actually a plate analysis is more correct,

especially because of the additional support of the front panel and possibly the tailgate. Par. 4-11 discusses more detailed calculations in design. Dump body walls are commonly reinforced at the floor-to-wall joint to reduce the magnitude of bending stress and also to act as a wear surface (par. 4-4.2). Beneficial results of reinforcement are clearly shown in Fig. 4-64(B); the maximum effective value of the bending moment $M'_{(max)}$ on the wall becomes

$$M'_{(max)} = \frac{\rho(h_1 - h_r)^3}{6}, \text{ in-lb-in. of wall length} \quad (4\text{-}78)$$

where

h_r = height of inside wall reinforcement measured from the floor, in.

Typical wall reinforcement details are shown in Figs. 4-64(C) and 4-54.

Sample Problem

Consider the 15-ton dump truck described in par. 4-5.1 and loaded as in Fig. 4-56. Find

a. maximum wall load intensity $q_{(max)}$

b. resultant pressure force acting on the wall F, and

c. the ratio of maximum effective bending moment in a reinforced wall to the maximum bending moment in the same, but unreinforced, wall $M'_{(max)}/M_{(max)}$.

Let h_r = 12 in.,

and ρ = 110 lb/ft^2 = 0.0637 lb/in.^{3}

a. $q_{(max)} = 34(0.0637) = 2.17$ lb/in.^{2}

This represents about 59 percent of the maximum floor load intensity under the heaped load.

b. From Eq. 4-76; $F = \dfrac{0.0637(34)^2(144)}{2}$

$= 5300$ lb

c. $$\frac{M'_{(max)}}{M_{(max)}} = \frac{(h_1 - h_r)^3}{h_1^3} = \left(1 - \frac{h_r}{h_1}\right)^3$$

$$= \left(1 - \frac{12}{34}\right)^3$$

$$= 0.271.$$

Therefore, from an approximate analysis, maximum effective wall bending moment was reduced about 73 percent by means of reinforcements.

Another category of cargo load that acts directly on walls is the load due to wall-hung equipment. Again, this wall load is limited to certain vehicle structures, most notably some of the van body types (par. 1-5.3). In shop and instrumentation vans, numerous electronic instruments, consoles, parts cabinets, and storage racks are fixed to the walls as well as to the floor structure[98,99]; the more delicate pieces of equipment are attached via shock mounts. While the wall is definitely placed under load, the magnitudes, normally, are not serious because of two factors—(1) the discrete wall mounting points can be placed at or near wall strength members, and (2) the floor will share a considerable part of the total load. A similar situation exists in combat vehicles, where numerous items such as ammunition, gas tanks, repair parts, and kits may be mounted to constitute a wall load. However, wall-mounted items represent a very negligible loading effect compared to the more serious limiting loads, road and ballistic impact, experienced by these vehicles.

Walls of vehicles can also be loaded indirectly by means of the cargo, i.e., the cargo need not be in direct contact with the walls. Indirect cargo loads occur in the more recent, lightweight design vehicles, where portions of the structure other than the floor, and primarily the walls, are purposely enlisted to carry loads by means of the stressed-skin technique. This type of design necessitates careful analysis of the load path from floor to walls (or other structural parts) to take advantage of the wall strength modes and minimize its weaknesses. This topic is discussed at length in par. 4-2.3.3, with particular emphasis on walls.

4-7.2 DYNAMIC WALL LOADS

Wall structures are subjected to many dynamically acting loads as are the other parts of automotive bodies and hulls. The difficulties of vigorous dynamic analysis and the lack of fully satisfactory design procedures are mentioned in connection with dynamic floor loads in par. 4-5.4. In that discussion, terrain- and cargo-imposed loads were pointed out as being particularly significant in design. Terrain-imposed load is equally important in the

design of walls because a load path is easily established between interconnected components of an automotive structure. The latter load source—cargo load—is not nearly as important in the design of walls as it is in floor design since direct wall loading occurs much less frequently than direct floor loading (par. 4-7.1). In general, the discussion and the load factor (or peak acceleration multiplier) type of analysis given for floor loads (par. 4-5.4) can also be applied to walls. Acceleration frequencies to be expected in cargo spaces of representative conditions are given in Tables 4-12 and 4-13 (par. 4-5.4). The portion of the data pertaining to longitudinal and lateral accelerations is applicable to wall structures.

A separate but common dynamic cargo load that acts primarily on wall structures is produced by centrifugal effects when a vehicle negotiates a curved course. The centrifugal acceleration gives rise to an acceleration force or cornering load that acts at the center of gravity of the cargo in a direction radially outward from the center of curvature. If the transported cargo is in direct contact with the wall, the cornering load component will substantially increase wall loading. It should be emphasized that the cornering load is a completely separate dynamic effect and is additive to the shock-type load developed between cargo and wall. (The latter dynamic wall load has been mentioned earlier in this subparagraph.) If the transported cargo is prevented from contacting the walls, cornering loads become important to floor and tiedown design.

Analysis of forces resulting from centrifugal acceleration is greatly simplified if a constant speed turn on a circular path is assumed. Such an assumption is not unreasonable in many automotive applications. For the preceding condition, tangential forces do not exist, hence it is sufficient to consider radial equilibrium only, from which

$$F = \left(\frac{W}{g}\right) R\omega^2 = \frac{Wv^2}{gR} \tag{4-79}$$

where

F = centrifugal force, lb

W = weight of cargo (or other object) undergoing centrifugal acceleration, lb

g = acceleration due to gravity, ft/sec^2

R = radius of curvature of path described by cargo CG about center of curvature, ft

ω = angular velocity of cargo (or other object) about center of curvature, rad/sec

v = constant speed of vehicle (or other object), fps

Some other dynamic loads that can have destructive effects on walls are (a) wind loads, (b) blast and ballistic impact loads, (c) loads encountered during airdrop operations, and (d) miscellaneous accidental impact loads. Load sources (a), (b), and (c) are included in a general discussion of dynamic loads (par. 4-1.2). Load (a) will be covered in greater detail in the paragraph which follows. Loads (b), (c), and (d) are of particularly destructive nature and only combat vehicles or specially constructed vehicles can stand up reasonably well agains them. Many military vehicles are so designed and constructed that permanent damage can result if they are exposed to load types (b), (c), and (d); the missions of these vehicles are, therefore, planned to eliminate such loads or minimize their chance of occurrence

Sample Problem

Let a high-speed, full-tracked, unarmored, amphibious cargo carrier with a payload capacity of 12,000 lb negotiate a 24-ft-radius turn on generally level terrain at a constant speed of 30 mph. The vehicle is on an ammunition transport mission, loaded with 100 2-round, standard, 105 mm ammunition boxes weighing 120 lb each. The cargo is arranged in two sections along the walls of a military cargo-type body, with a small longitudinal aisle left between the sections. Find (a) the centrifugal force exerted on the wall outboard to the turn, and (b) the ratio of the dynamic load on the wall to the static load on the floor produced by the accelerated cargo.

(a) The effective load on the wall outboard to the turn is one-half of the total payload because of the cargo arrangement, or W = 6,000 lb. The center of gravity of the effective load will have a radius of curvature somewhat smaller than the vehicle turn radius; assume

R = 22 ft, v = 30 mph - 44 fps, then

$$F = \frac{6000(44)^2}{32.2(22)} = 16{,}400 \text{ lb}$$

(b) $\frac{F}{W} = \frac{v^2}{gR} = \frac{(44)^2}{32.2(22)} = 2.74$

Note that this cornering load would act on the wall in addition to any shock type dynamic load developed between cargo and wall.

In the preceding example, only one-half of the cargo load is assumed to act on the wall; and the ratio of dynamic to static load (part b) is based upon one-half of the total cargo load. If the cargo had been arranged in a solid book, would equal 12,000 lb, the centrifugal force would be doubled, but the ratio would remain the same.

4-7.3 WIND LOADS ON WALLS

Consideration of wind loads plays a relatively small part in the design of military automotive structures. This is not due to any lack of potentially destructive strength possessed by high winds; but because military vehicles are regularly exposed to other severe sources of loads, and the considerations necessary to resist some of these are sufficient against wind effects also. The frequency of high wind conditions, wind velocities, (hence, geographical locality), height, shape, and size of the structure concerned are factors that must be weighed to determine the value of wind load analysis. The low height and massive construction of many military vehicles offset their poor aerodynamic shape and often obviate consideration of wind loads. It is the lightweight type of vehicle construction that is most susceptible to wind loading. In particular, van bodies (par. 1.5.2) with sizable wall surface areas suffer the most from the shape and size factors. Although vans are relatively tall structures automotively speaking, it is in smokestacks, towers, and bridge girders that the height factor becomes important.

The type of damage that high winds may cause in vehicles is overturn, from dynamic pressure acting laterally against a wall, and roof lift or wall blowout, from unequal pressures on either side of a member. The latter effects are caused by nonuniform flow of air around the structure. Structural failure can also result from self-excited or forced vibrations induced into the structure by gusts or flow oscillations if their frequency approaches that of the structure's natural frequency.

Officially available wind velocities are either a *maximum* or an *extreme wind velocity*[22]; the former is an average velocity measured over five minutes, while the latter is the average velocity over the time taken to cover a distance of one mile. Such standard but long duration wind measurements are not adequate for wind load-strength calculations. To compute the dynamic pressure on a wall, the *critical maximum velocity* is needed which is the maximum velocity of gusts comparable in size with a given structure; e. g., for a 60-mph wind lateral to an 8-ft wide vehicle, it would be a gust of only 0.09 sec in duration. Specialized, very-low-inertia instruments can measure instantaneous velocities of short-duration wind gusts, and test results have shown that standard wind velocity measurements can be modified to give actual wind velocities by means of a *gust factor*. Typical values of the gust factor range from 1.2 to 1.5 (Ref. 100). The dynamic pressure P_d due to wind load on a wall is given by

$$P_d = \frac{\rho (kv)^2}{2}\text{, psf} \tag{4-80}$$

or,

$$P_d = \frac{\gamma (kv)^2}{9260}\text{, psi} \tag{4-81}$$

where

ρ = air density, slug/ft^3 (0.00237 slug/ft^3 at 60°F and standard atmospheric pressure)

γ = air density, lb/ft^3 (0.0764 lb/ft^3 at the same temperature and pressure as for ρ)

v = wind velocity, fps

k = gust factor, dimensionless

Refs. 101 and 102 prescribe basic wind conditions that military vehicles are expected to meet. These conditions are:

(a) winds of 45 knots* for a five-minute duration, with gusts of 65 knots, and

(b) with hold-down equipment, 55-knot winds of five-minute duration and 85-knot gusts.

Duration of gusts for conditions (a) and (b) is not given; but, for convenience, gust velocity divided by five-minute velocity can be taken as a

*1 knot = 1.151 statute mile per hr.

gust factor. This result in values of 1.44 and 1.55 for cases (a) and (b), respectively. It should be kept in mind that higher gusts in the critical, short-duration range are possible. In any case, conditions (a) and (b) refer to basic, inland geographical areas, where military vehicles can be expected to operate most of the time. Substantially higher winds can be experienced in mountainous and seashore areas[101].

Sample Problem

A parked, 2½-ton van truck is struck at right angles to its longitudinal centerline by sustained winds of 55 knots accompanied by 85-knot gusts. Assume atmospheric conditions are: standard pressure and 60°F. Find the dynamic wind load intensity on the windward wall of the van body

$$v = 55 \text{ knots} = 1.151(55) \text{ mph} = 93 \text{ fps}$$

$$k = 1.55$$

then

$$P_d = \frac{0.0764[(1.55)(93)]^2}{9260} = 0.171 \text{ psi}$$

$$= 24.6 \text{ psf}$$

The wind load in this example is high, but it is not an extreme load condition when compared to other expected vehicle loads discussed in other sections of this handbook. The wind conditions used in the preceding sample problem correspond to the more severe of two basic wind conditions given in Ref. 101. Therefore, it can be generally concluded that wind loads from natural causes are not detrimental to vehicle structures except perhaps in isolated, extreme cases encountered in seashore areas during hurricane conditions, or in some mountainous regions. Another extreme condition exists in the possibility that land vehicles can be exposed to the fantastically high velocity, "unnatural" winds created by the detonation of nuclear weapons. Aside from thermal and radiation damage, the destructive forces of a nuclear detonation are blast and shock effects. The specific damage agents are (a) the peak overpressure due to the shock wave, and (b) the peak dynamic pressure due to the blast winds created. Both types of pressure loading act simultaneously but one or the other component will be more important to a given structure. Land transportation equipment has been found sensitive to dynamic pressures rather than overpressures[103].

Table 4-15 shows the relation among peak overpressure, peak dynamic pressure, and maximum wind velocity for an ideal shock wave traveling in air at sea level. The data in the second and third columns can be used to picture the ultimate magnitudes of wind loading and relate these to the more familiar, natural magnitudes. The bottom two rows of information are readily imaginable and relatable to natural occurrences. The wind velocity and dynamic pressure found in the sample problem of this paragraph would fit just above the bottom row of data. Note that this wind condition is specified as a climatic design criterion for most vehicle operation areas[101]. Somewhat above the last two rows of data in Table 4-15 the limit of natural peak winds is reached in the form of unusually severe hurricanes and typhoons. Higher wind velocities become difficult to imagine but would be found at various distances from the point of detonation of high explosives and nuclear weapons, depending on weapon size. The top row of figures represents wind conditions near ground zero or the *hypocenter* of small-yield nuclear bombs.

The wind velocity and dynamic pressure that correspond to the onset of detrimental structural damage in a vehicle are hard to specify

TABLE 4-15 OVERPRESSURE, DYNAMIC PRESSURE, AND WIND VELOCITY IN AIR AT SEA LEVEL FOR AN IDEAL SHOCK FRONT[104]

Peak Overpressure, psi	*Peak Dynamic Pressure,* psi	*Maximum Wind Velocity,* mph
200	330	2,080
150	223	1,778
100	123	1,414
72	80	1,170
50	40	940
30	16	670
20	8	470
10	2	290
5	0.7	160
2	0.1	70

because of several influencing factors. These include the degree of protection offered by the topography, vehicle orientation with respect to the wind, and the ability of the structure to absorb damage and remain functional with little or no corrective effort. The term "little corrective effort" includes such actions as righting of upset vehicles, the cutting away of damaged but expendable components, etc.

Data available from nuclear bomb effects on Japan and from tests in this country during the 1950's verify that land transportation vehicles can survive substantial blast-wind loading. The types of vehicles tested that are of interest here include passenger automobiles, busses, trailers, mobile homes, emergency vehicles, and earth-moving equipment. Ref. 103 concludes that emergency vehicles can be expected to remain functional after exposure to a blast overpressure of approximately 5 psi and an associated dynamic pressure of about 0.7 psi at wind velocity of 160 mph (Table 4-12). Vehicles of lighter construction (passenger cars, vans, mobile homes) were shown to be less resistant to wind loads. Note that these wind conditions are substantially above the climatic design criteria given in Ref. 101. Typical damage incurred by the emergency vehicles included broken glass, dented and distorted sheet metal, and overturn when the vehicle was oriented broadside to the wind. Since emergency vehicles are somewhat comparable to military transport vehicles, the 0.7 psi wind load intensity figure can serve as a rough overall indication of detrimental damage threshold. Transport vehicles of heavier construction and combat vehicles can sustain much higher loads, however.

Additional concern for wind loads on land vehicle structures comes from consideration of resistance to motion. Aerodynamic losses as related to military vehicle design are discussed briefly in par. 4-6. Other design guidance in this general area is given in par. 4-1.3.4.

4-7.4 HYDROSTATIC LOADS ON WALLS

The action of hydrostatic loads on wall structures has been briefly discussed in connection with inclined and curved floors which are, essentially, walls. For the analysis of hydrostatic loads on vertical and inclined walls, Eqs. 4-75 to 4-77 of par. 4-7.1 are readily applicable. These equations are primarily for hydrostatic loads, but were presented originally for wall loads resulting from constant-density loose bulk cargo which also obey laws of hydrostatics. With actual hydrostatic loading, the direction of load would, of course, be reversed to that in par. 4-7.1. The equations are rewritten here for convenience, with terms more suited to amphibious vehicle analysis. Eqs. 4-82 (load intensity) and 4-83 (pressure force) may be used for all types of wall construction, while Eq. 4-84 (bending moment) is limited to approximate calculations for military cargo body types of walls (par. 4-7.1).

$$q_{(max)} = \rho h'_{(max)},\ \text{lb/in.}^2$$
$$= \frac{h'_{(max)}}{27}\ \text{(for seawater)} \tag{4-82}$$

$$F = \frac{\rho\, [h'_{(max)}]^2 \ell}{2},\ \text{lb}$$
$$= \frac{[h'_{(max)}]^2 \ell}{54}\ \text{(for seawater)} \tag{4-83}$$

$$M_{(max)} = \frac{\rho [h'_{(max)}]^3}{6},\ \text{in.-lb/in., of wall length}$$
$$= \frac{[h'_{(max)}]^3}{162}\ \text{(for seawater)} \tag{4-84}$$

where

$h'_{(max)}$ = maximum immersion depth of the wall, into a fluid medium, in.

ρ = density of the fluid medium, lb/in.^{3}

ℓ = length of the wall, in.

$M_{(max)}$ = maximum bending moment, in.-lb/in. of wall length.

In cases where portions of a wall structure are inclined as shown in Fig. 4-65, the linear variation of hydrostatic load intensity with immersion depth remains unchanged. Orientation of the load intensity, pressure force, and pressure prism will be perpendicular to inclined surfaces.

q = Load Intensity, lb/in.^{2}

F = Resultant Pressure Force, lb

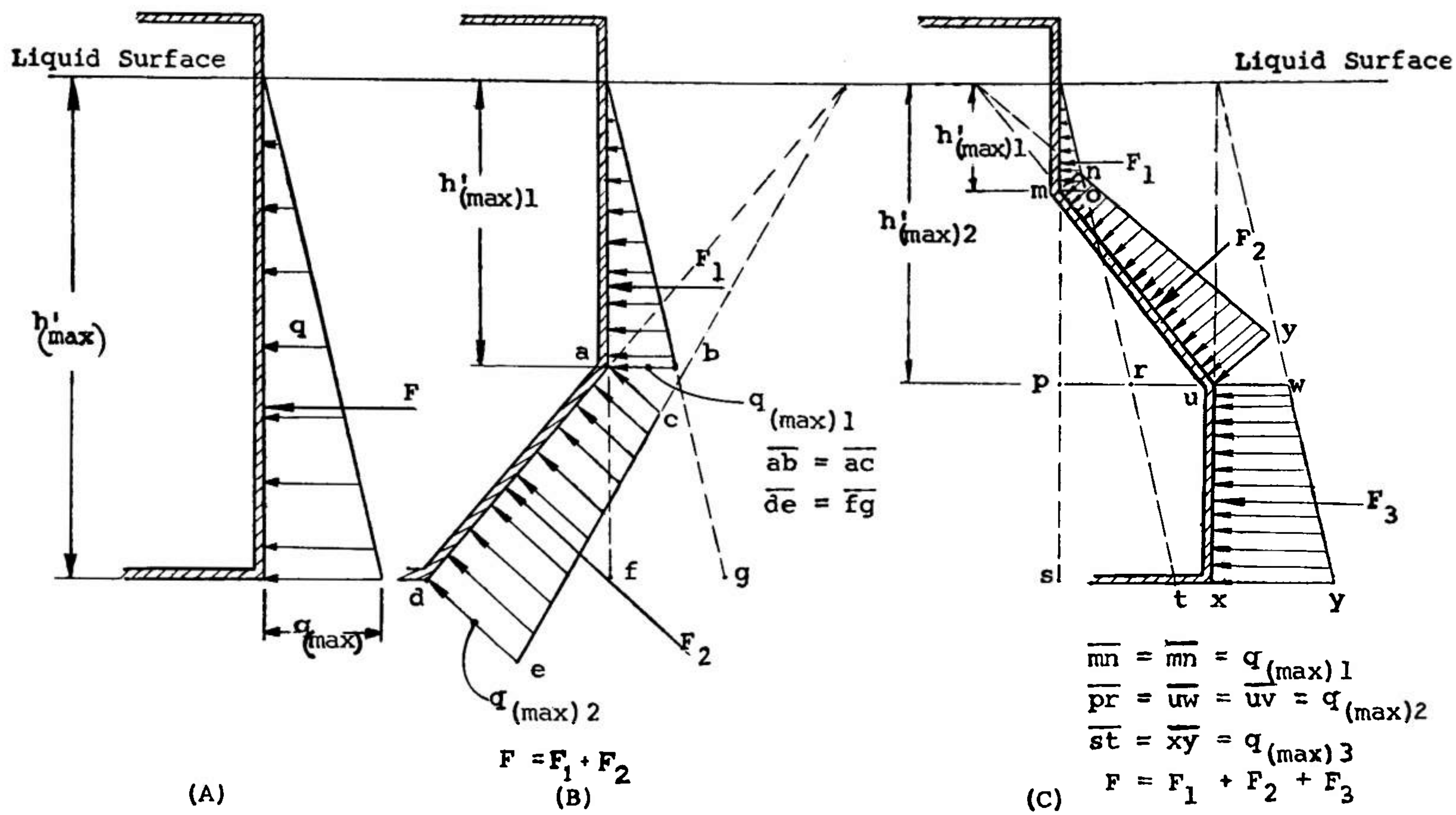


Figure 4-65. Hydrostatic Load Comparison Between Vertical and Inclined Walls.

SECTION IV—ROOFS—CONSTRUCTION AND DESIGN

4-8 GENERAL CONSIDERATIONS

Permanent roofs in noncombat vehicles are found only on van bodies and in body-on-chassis construction (Figs. 1-11 and 1-3, respectively). A particular exception is the tank-type body which does indeed exhibit a roof, but the overall construction of this type body is homogeneous to the degree that floor, wall, and roof design cannot be logically separated. For this reason the construction and design of tank-type bodies are treated separately in Section VI of this chapter. (See Sections I and II of Chapter 1 for detailed physical characteristics of wheeled vehicle body types and tracked vehicle hull types, respectively.)

Combat vehicles almost always have permanent roofs due to the nature of their intended combat missions. Therefore, the construction details and design of roofs in combat vehicles are largely similar to the floor and wall construction described in Sections II and III of this chapter. However, differences to be noted for general considerations are:

a. The material thickness needed is a function of the degree of the threat against the structure surface—roofs generally experience a larger threat than do walls, while the threat from mines against floors is quite considerable.

b. The presence of a significant number of hatches, doors, access openings, or grilles on roof surfaces—in contrast to walls. Because of greater homogeneity in combat vehicle structures, the following paragraphs deal mainly with roof details of noncombat, van-type vehicles.

Roofs must be constructed so as to be compatible with the rest of the automotive structure. The result is that even in noncombat vehicles certain similarities between various parts of the body exist in the structural components and material gages used. For example, Military Specifications for some vehicles with van-type bodies call for the identical material gage specified as minimum thickness for wall and roof outer panels (see van body references, Table 4-14).

Normally, roofs are lightly loaded members, but strength for ordinary loading is not the only consideration. Roofs, like walls, are exposed to accidental punctures, dents, and impacts. These highly concentrated sources of load can arise either exterior to the vehicle, as from impacts with overhead obstructions, or interior to the vehicle, as from accidental damage by cargo-handling equipment. Minimum roof gages specified often include allowance for limiting these types of accidental structural damage.

A number of transport vehicles, primarily wheeled but also some of the track-laying variety, have semipermanent roofs of fabric material. The semipermanent roof is a rubber-impregnated, moisture-resistant, tarpaulin stretched over supporting roof bows positioned at intervals along the sides of the cargo body. When not needed, the tarpaulin can be removed and stowed aboard the vehicle thereby facilitating cargo handling. The roof bows are also removable; when unused they are positioned out of the way at the extreme forward end of the body as shown in Figs. 1-6 and 4-61. Figs. 1-7 and 1-8 illustrate typical semipermanent roofs in use. The function of the semipermanent roof is mainly to provide a measure of protection for cargo and personnel from the weather; however, it has been used occasionally for other purposes, e. g. as a form of camouflage on fuel tank trucks.

4-9 TYPES OF ROOFS

The automotive roof structure is a combination of logically placed strength members and sheet metal panels. The strength members are roof rails, rub rails, purlins, and permanent roof bows (compare to roof bows in par. 4-8) which are formed by rolling or extruding techniques. Permanent roof bows are laterally oriented on the vehicle for roof support. They are generally produced of lightweight aluminum castings[73]. Roof rails and rub rails are positioned either longitudinally or laterally to the vehicle. These members provide the main support for the roof panels as well as integrate the roof structure into the wall structure (Fig. 4-66). Purlins act as auxiliary longitudinal roof supports but are not used in all applications. The construction of busses employs roof purlins.

Figs. 4-58 to 4-60 show, in part, typical

examples of automotive roof construction and also illustrate some of the foregoing nomenclature. Lateral cross sections of the bodies shown in these figures provide additional useful information about roof types and their relation to other parts of the body. Fig. 4-66 shows typical van body cross sections in which roof and rub rails are roll-formed. In Fig. 4-66(A), the roof rail shape includes provision for a gutter and for mating with lateral roof support members. Here these members are square tubes very similar to those used for the wall strength members. A similar roof rail is used in combination with one type of roof bow shown in Fig. 4-66(C); the small offset near the inboard edge of the rail is used to position the roof paneling itself. A simpler roof rail form used with crowned roof bows appears in Fig. 4-66(B). Still another construction type (Fig. 4-66(D), uses a rub rail (which is primarily a wall structure member, see par. 4-6.1) as the main roof support along with square tube lateral members. A lintel member is used in all four types of construction described to tie the upper wall structure to the roof rails. With steel construction, it is necessary that the lintel and roof rail be separate components, since roll-forming is limited to noncomplex shapes.

With aluminum construction, the highly developed manufacturing process of extrusion forming makes roof rails of complex, one-piece sections readily available. This process makes it possible to include gutter, lintel, and even rub rail features into one-piece roof rail sections (Fig. 4-67(B), (D), and (E)) and, to provide full length sections for most body lengths. Fig. 4-67 gives typical aluminum roof construction details; the material thickness of the roof rails varies from 0.1 to 0.24 in. Fig. 4-67(A) shows the complete roof rail as composed of two pieces; this arrangement permits the wall and roof to be built separately, with the latter assembled to the former as a complete unit. In Fig. 4-67(C), an extra lightweight, two-piece roof rail is used along with top-hat sectioned roof bows bent to shape. The lower roof rail component contains a gutter molding; and, again, the complete roof structure can be attached to the wall as a separate unit[106].

If roof-to-wall curvature is neglected, the constructional examples in the first three parts of Fig. 4-67 have a flat-roof configuration which has less rigidity than the crowned roof illustrated in Fig. 4-66(B) and (C). Fig. 4-67(D) and (E) also show construction wherein the roof bows are cambered to develop additional strength from roof panels, as compared to flat panels of equal thickness. Fig. 4-67(D) shows an especially light roof rail section, while Fig. 4-67(E), indicative of heavier duty use, has a more substantial roof rail section and is additionally reinforced by an inner rail section plus gussets between wall supports and the roof rail. The wall posts and roof bows illustrated in Fig. 4-67 are top-hat sections exclusively—naturally other structurally efficient shapes (par. 4-2.2) can be used equally well.

Permanent or structural roof bows, contrasted to the semipermanent bows discussed in the previous paragraph, are very important components of the total roof structure. They unite the two roof rails located at the longitudinal edges of the roof and give the roof panels their characteristic shape. Types of roof bows common to van body design are shown in Figs. 4-68 and 4-69; all three of the examples feature roll-formed sections bent to the desired shape. Material gages and pertinent dimensions shown are typical of these members. Fig. 4-68 illustrates two nearly identical roof bows; the second example is reinforced by an additional member to produce an advantageous closed cross section. A roof bow composed of three parts with identical sections and joined in two places is shown in Fig. 4-69. The center portion of this bow has a small but finite curvature (large radius of curvature) so that the roof panel placed over it will conform to this shape and thereby derive extra strength. Roof bows are often somewhat overcrowned in order that a tight fit be obtained between them and the roof panel. Rubber friction pads are also employed at this location.

The roof panel itself is sheet steel or sheet aluminum material of 20-gage thickness (see par. 4-8). Panels may be affixed with relative ease on well prepared understructures, such as shown in Fig. 4-58 and in the details of Fig. 4-67 for slightly different construction. The use of light but sturdy roof bow sections and well designed roof rail shapes has been instrumental in easing and simplifying installation of roof panels. Certain roof rail shapes (Figs. 4-66(C) and 4-67(B)) include a useful recess near the inboard edge which helps to position panels laterally and provides a convenient place for longitudinal edge sealing.

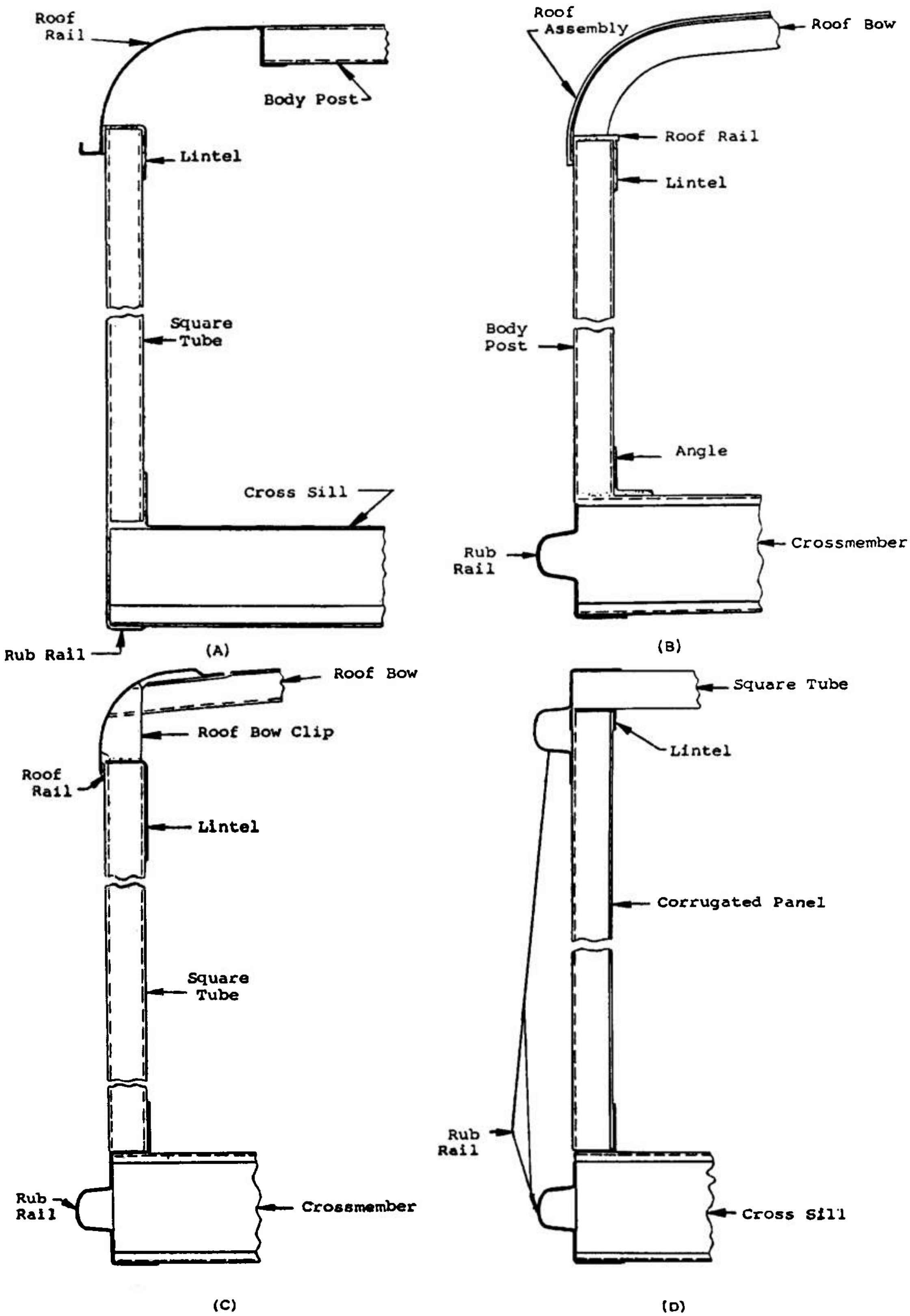


Figure 4-66. Typical Roof and Wall Construction[105]

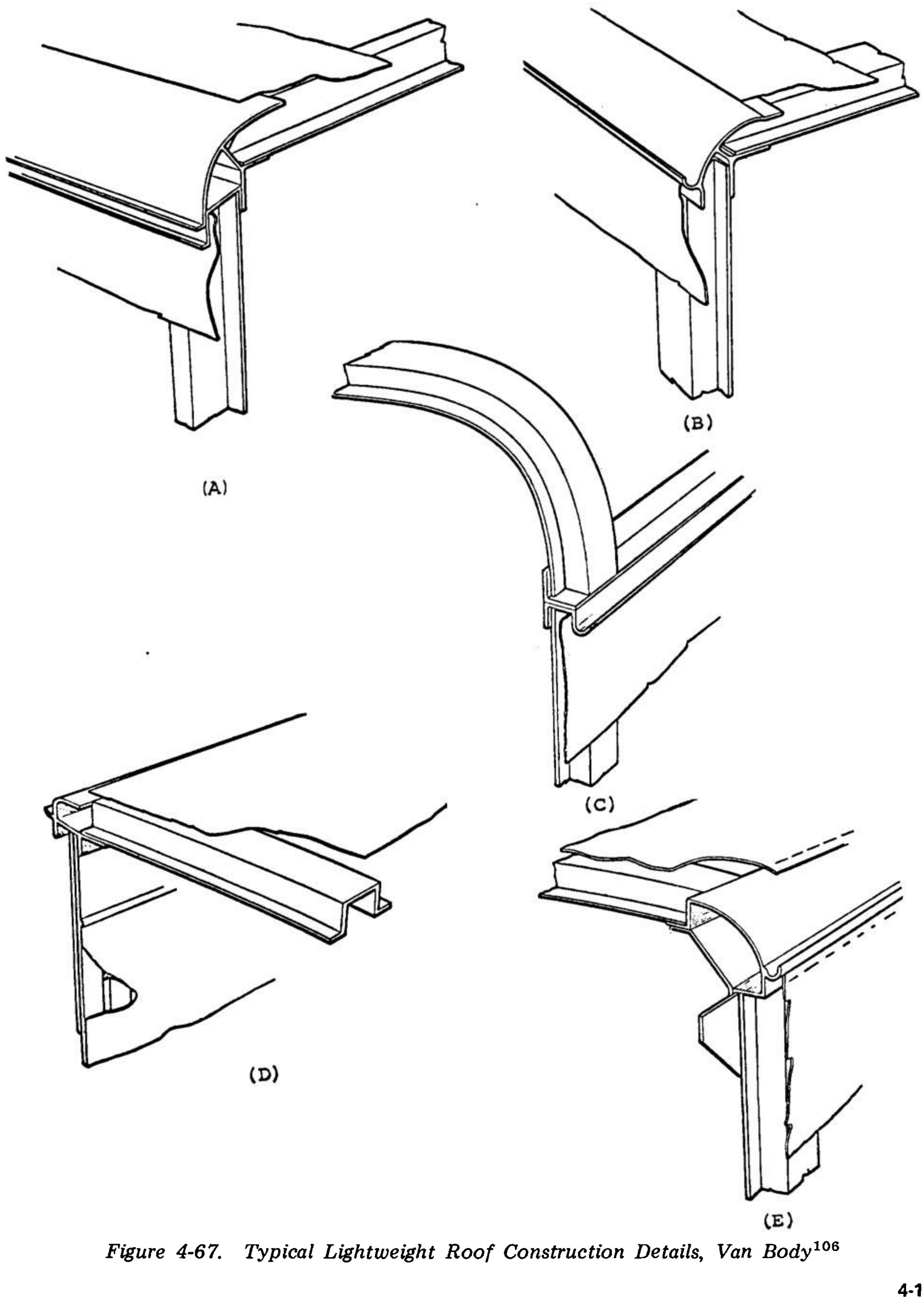
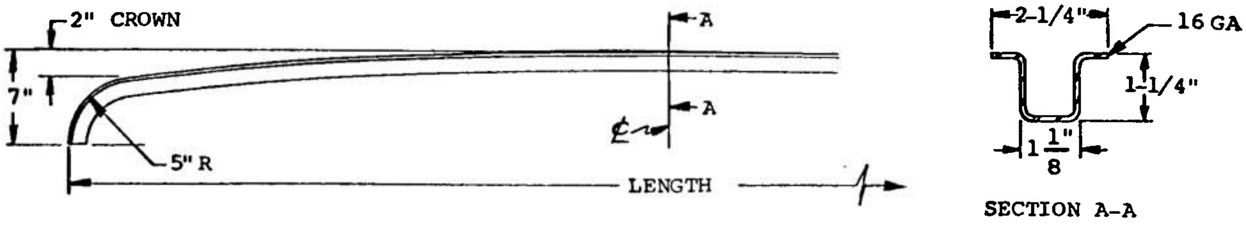


Figure 4-67. Typical Lightweight Roof Construction Details, Van Body[106]



Typical Standard Lengths: 82, 84, 86, 88, 90, or 93-1/4 in.

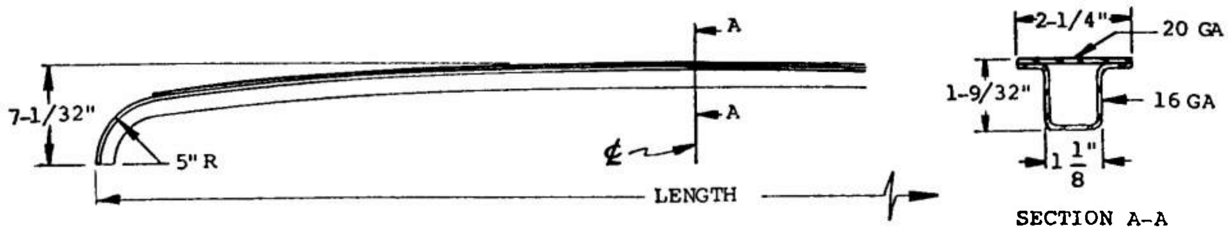


Figure 4-68. Roof Bows[107]

It is most convenient to position roof panels transversely to the body, i.e., to cut available sheet metal stock widths to the width of the roof panel required and join them with transverse seam to obtain the required roof length. The construction types outlined in this paragraph give wide latitude to the vehicle designer in the placement and pitch of roof supports. (The same is true for wall supports, but the respective pitches of the former and the latter can be independent of each other.) If roof panels are lap-jointed, certain roof bows should coincide with the joint location. Ref. 106 recommends that the pitch spacing of lap joints be double that of the roof bows. Numerous other joints are used on roof panels; one of these is the cap-strip sheet-metal joint illustrated in Fig. 4-70. The joint location need not necessarily coincide with the location of a roof bow when this joint is used. In Ref. 107, the cap-strip joint is used on 24- or 36-in. centers (as an economic standard) to produce complete roof panel assemblies of required width and length. This is the kind of assembly referred to in Fig. 4-66(B).

Many constructional details are encountered in connection with roof structures, as can readily be seen from Figs. 4-58 to 4-60. Some of these details have been discussed and illustrated in this paragraph; additional details are given in Fig. 4-71. Fig. 4-71(A) shows the construction at any one of the four roof corners. Identical extruded aluminum roof rail and wall pillar sections (mitered at the common corners) are used along with separate, inner reinforcement sections. The roof panel is lap-jointed and capped by a special corner roof piece. The same corner detail, but without the reinforcing sections, is shown in Fig. 4-71(B) as viewed from inside the body. Rigid connection between the three mutually perpendicular structural members is obtained by use of gussets. Fig. 4-71(C) gives an inside view of a typical roof-to-wall connection away from the roof corner. In particular, a roof bow to roof rail to wall post joint is shown; the roof bow and wall post are identical Z-section members.

4-10 ROOF LOADS

The specification of roof loads is dependent upon vehicle classification at least as far as the combat and tactical vehicle categories. Types and magnitudes of roof loads for the former class are not basically different than the floor and wall loads discussed in pars. 4-5 and 4-7, because of the rather homogeneous overall construction required for those vehicles. Considerations that apply specifically to roofs are given in par. 4-8.

In tactical vehicles the roof structure, if any, is a comparatively lightly loaded member; although both direct and indirect loads act upon it. Snow loads and infrequent loads caused by

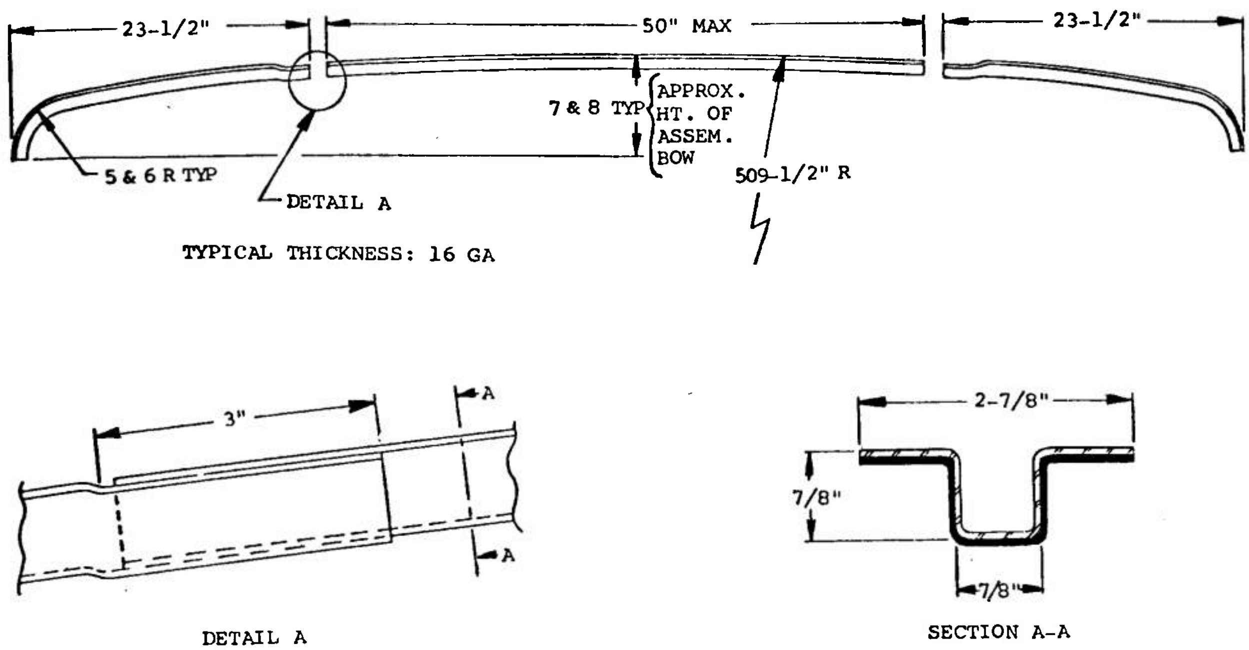


Figure 4-69. Additional Roof Bow[107]

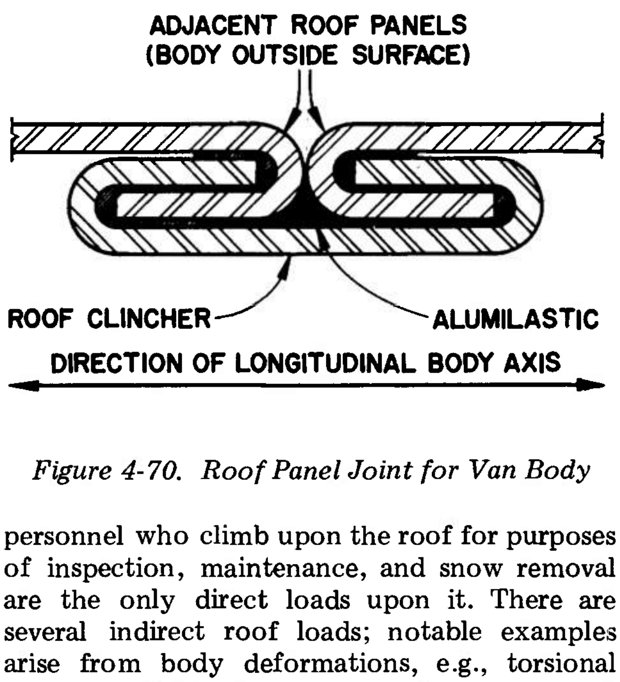


Figure 4-70. Roof Panel Joint for Van Body

personnel who climb upon the roof for purposes of inspection, maintenance, and snow removal are the only direct loads upon it. There are several indirect roof loads; notable examples arise from body deformations, e.g., torsional loads (par. 4-5.4) due to terrain irregularities and lateral bending due to cornering (par. 4-7.2). Snow and personnel loads appear to be insignificant relative to the indirect loads which, however, are the most difficult to evaluate quantitatively.

Snow load has to be regarded as a marginal vehicle loading effect because of its intermittent occurrence in most geographical regions. Ref. 101 describes eight climatic regions that range from intermediate to the extremes of hot and cold, and into which all land areas of the world may be classified. Only the intermediate and the cold climatic categories experience snowfall; and for these, design snow loads have been specified[172] and are presented in Table 4-16. Snow loads are functions of terrain altitude and frequency of snow removal (which, in turn, are measures of vehicle mobility or equipment portability). For example, the temporary designation in Table 4-16 is representative of tactical vehicles that are moved often and cleared of snow between storms; the semipermanent classification is applicable to instrumentation or shop van trailers parked for long intervals. The right-hand column of the table expresses snow load intensity in units of lb/in.^{2} for conveninet comparison with loads examined in preceding sections of this chapter. Such a comparison shows snow loads to be rather light and close to wind load values (par. 4-7.3). Table 4-16 agrees reasonably well with the snow load criterion for buildings, for which Ref. 108 lists a value of "not less than 30 lb per horizontal sq ft of roof".

It is interesting to look at an average snow depth represented by, for example, a loading of

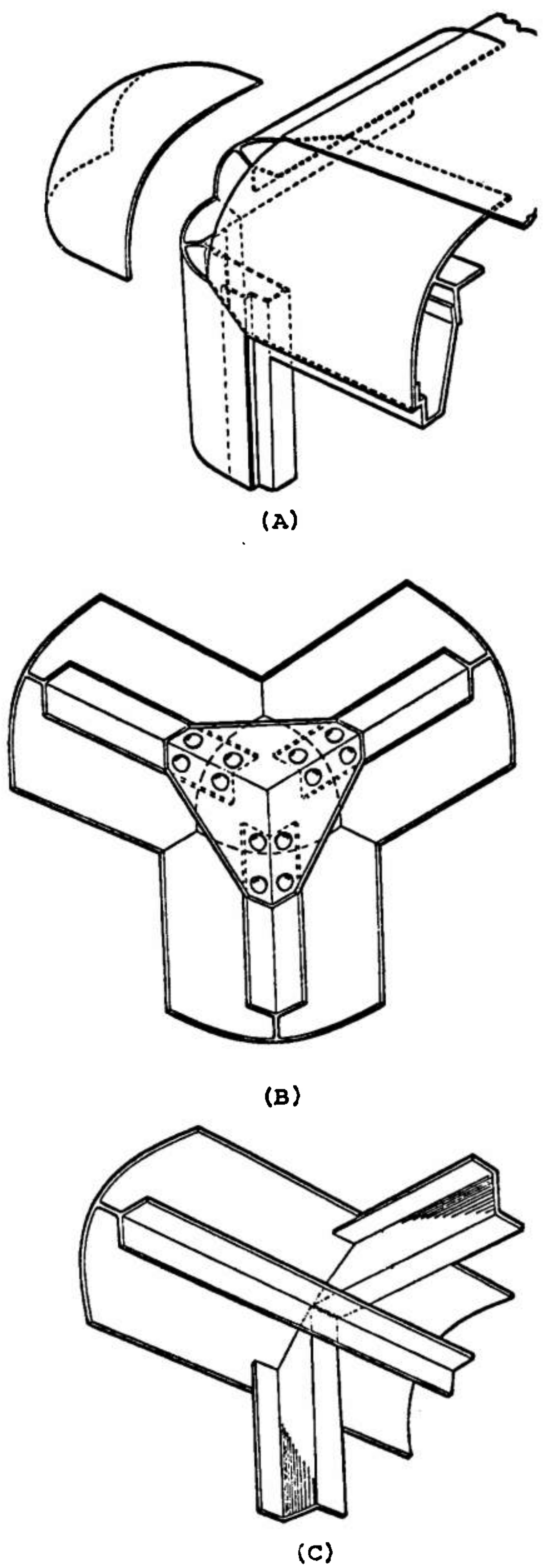


Figure 4-71. Roof Construction Details[106]

TABLE 4-16 SNOW LOADS

	SNOW LOADS			
	lb/ft²		lb/in.²	
	Altitude, ft		Altitude, ft	
	<4000	>4000	<4000	>4000
Temporarily installed equipment, or parked vehicle (periodic snow removal)	20	25	0.139	0.174
Semipermanently installed equipment, or parked vehicle (snow left to accumulate)	40	50	0.278	0.348

30 lb/ft^3. If Eq. 4-67 is used to determine the snow depth, and a value of ρ=8 lb/ft^3 is used for the density of new fallen snow (Table 4-11),

$$h = \frac{q}{\rho} = \frac{30}{8} = 3.75\ \text{ft} = 45\ \text{in. (snow depth)}$$

A typical snow depth is hard to predict because there are many types of snow with greatly varying densities and also because of wind pile-up effects. The snow density used in the example has a specific gravity of 0.128 which compares well with a value of 0.15 given in Ref. 109 for one grain form and size of new snow. However, other *in situ* specific gravities as low as 0.03 for wild new snow and up to 0.7 for rotten spring snow are also listed.

The torsional strength of an automotive body with a roof is difficult to estimate; specific design guidance is lacking. Certainly, the closed, rectangular, tube-like cross section completed by the roof substantially increases this strength. But the lighter construction that is representative of noncombat vehicles demands that methods other than mere sectional strength be employed to counteract racking and twisting of the body imposed by terrain. Some of these other considerations are increased structural flexibility of the body by increased, but controlled, deflection in members; use of flexible joints between members (e.g., rubber friction pads between roof bows and panel); improved load distribution techniques; improved suspension systems; and new construction concepts such as the articulated tractor-carrier combination (Fig. 4-57).

Lateral bending of a long body when cornering, and associated centrifugal loads that are due to the cargo also apply loads on a roof

structure. Although the vehicle floor offers the primary restraint on such forces, it is obvious that the roof can resist sizable forces and moments acting in its own plane. Efficient automotive structures utilize the beam strength of the roof assembly in much the same manner as wall beam strength is used to resist floor loading (par. 4-2.3.3). In the latter case, direct or normal floor loads are transferred and become more compatible loads in the plane of the wall. In the former case, bending and centrifugal loads normal to walls are transmitted to the roof, and the analytical methods of par. 4-2.3.3 can be extended to the analysis of roofs.

4-11 COMMENTS ON DETAILED CALCULATIONS OF FLOORS, WALLS, AND ROOFS

The design calculations presented throughout this chapter have been carried out primarily to the point of specifying load magnitudes and sources only. Bending moment and stress level calculations have not been performed except for specific examples of lightweight construction techniques—the honeycomb sandwich (par. 4-2.3.2) and the stressed skin structure (par. 4-2.3.3).

There are several reasons for adopting this procedure:

a. Sources and magnitudes of vehicle loads are the most important data to the designer, and usually not sufficiently known.

b. More detailed calculations require that very specific body constructional details be assumed in a vehicle. This is contrary to the purpose of a general handbook.

c. Detailed calculations are, in general, well known to the design engineer, once the load data have been obtained.

In spite of the many kinds of construction to be found in automotive bodies and hulls, they all are more-or-less complex systems of beam-like and plate-like structural combinations. Given the applied loads, methods of solution for bending moments, stress levels, and deflections are thoroughly described in numerous texts, e.g., Refs. 1 through 6 and 112 through 120.

In the case of plate-like members, the two-dimensional state of stress requires that sets of plate coefficients be used to determine bending stresses, deflections, and buckling loads. The difficulty involved with these coefficients centers around the need to specify, or to assume, the plate edge support condition. This problem is discussed in greater detail earlier in the chapter. These coefficients are functions of the plate dimension ratio and edge support condition for a given load on a member. Different coefficients are used for bending moments, deflections, and buckling loads. Thus, although a considerable number of coefficients might be of interest to the design engineer, for practical reasons only an illustrative sample is presented. Refs. 116 to 118 treat this area in great detail Ref. 118 is especially thorough in its presentation of buckling in structures.

SECTION V—JOINTS—CONSTRUCTION AND DESIGN

4-12 JOINT REQUIREMENTS

Because joints in automotive structures comprise a vast topic, treatment in a limited space necessitates that discussion be confined to specific areas. The references cited in this section should be consulted for additional details.

The joints with which the vehicle designer is most concerned are made by bolted, riveted, and welded connections between members. From the strictly military viewpoint, bolting and riveting have a distinct disadvantage—bolt- and rivet-type fasteners are vulnerable to ballistic impact and become dangerous projectiles themselves inside the vehicle. Welded joints are not completely free from danger of ballistic attack—in the form of fluidized welds—but proper design procedures (par. 2-13) minimize these effects. Therefore, welding is used extensively for joint fabrication in combat vehicles. The watertight quality of well-produced, continuous seam welding has also made it attractive to amphibious vehicle applications. Riveting is also used widely on tactical vehicle bodies; bolting is less popular since greater attention is needed to ensure joint tightness. Nevertheless, bolted joints are found in military vehicle structures—often on disconnectable and access joints (e.g., the cab

protector on a dump body, par. 1-5.3).

On the other hand, riveting enjoys certain advantages over bolting and welding. Riveting does not need auxiliary components or time-consuming installation as bolting does, nor does it require the special inspection techniques to assure reliability that welding does. Moreover, in a cold riveting application, the joint material does not suffer strength reduction due to heat as is the case with welding. Thus, cold-riveted structures can be composed of lighter sections because anneal compensation is not necessary. Of course, riveting has relative disadvantages, as does any joining technique; and final process selection is usually based on a combination of specific vehicle requirements and engineering trade-offs.

Requirements of structural joints include sealing, watertightness, compatibility of fastener and joint materials, and fastener arrangement to minimize eccentric loading. The sealing requirements for joints were discussed in par. 3-7 in connection with weather sealing. It suffices merely to add that this seemingly elementary requirement is a significant problem area in van body construction.

Joints in automotive structures are very often subjected to eccentrically applied loads. The joint is more severely stressed because the action of moment load in addition to the direct load is inherent in such cases. While load eccentricity cannot be eliminated, the vehicle designer can minimize its effects by proper arrangement of fastener or weld groups with respect to the load. Efficient fastener- and weld positioning must start with a joint load analysis. Eccentric loads on joints are treated in greater detail in par. 4-14. The other joint requirements previously listed are covered in the paragraphs dealing with the specific fastening methods used.

4-12.1 BOLTED JOINTS

The most basic requirement in any application is that joint integrity be maintained under applied loads. For bolted joints this means that all fasteners must be correctly tightened by bolt tension. Excessive bolt tension can cause bolt failures or panel yielding, while insufficient tension will loosen components and eventually lead to joint separation. The accepted method for establishing bolt tension in joints is to specify assembly torque values for the bolt heads or nuts. While this method is very convenient, the influence of uncertain friction between the surfaces of the bolt and the bolted part can cause serious overstressing or understressing of components. Improved bolt tensioning techniques, magnitudes and types of bolt loads, and some specific bolt torque-load recommendations are given in par. 4-14.

Coarse-threaded fasteners are preferable to the fine-thread series since there is little need for the fine adjustment advantage of the latter in structural work. Larger fasteners are more convenient from the installation and strength standpoint[119]. Threaded fasteners should also be limited to a small number of preferred diameters, lengths, and materials. Ref. 119 also lists a number of useful military and industrial specifications on threaded fasteners and rivets.

Since many automotive applications of bolted joints are of the disconnectable type, sufficient clearance around bolt heads and nuts must be allowed for tool clearance. Refs. 120 and 121 give bolt clearances required for hand assembly tools (open end, box, socket wrenches), and power tools (bolt impact wrenches), respectively, The bolted joint in aluminum construction requires additional attention to detail. Mild steel, stainless steel, and aluminum alloy bolts are all used to fasten aluminum parts[73,122]. Effect of corrosion due to electrolysis between different materials and appropriate methods of control (par. 2-9) should not be overlooked by the design engineer. Tight fit between bolt and bolt hole is important. A slight interference is preferable, according to Ref. 122, while Ref. 73 recommends reamed holes of minimum clearance. When such close tolerances are not practical, any class of fit may be used if a reduced bolt strength is used in the design calculation for the number of bolts needed. Of course, the use of aluminum alloy bolts necessitates lower allowable bolt stresses and, hence, a somewhat greater number of fasteners.

Lock nuts and spring washers should be used in conjunction with threaded fasteners, in general, to reduce joint loosening under repeated loading. It should be noted, however, that the commonly used bolt torque-load relations do not take into account the presence of supplementary locking devices (par. 4-14). For aluminum structural members, use of plain washers under all nuts, bolt heads, and especially spring washers is recommended[73,112] to increase

the effective bearing area. The foregoing statement is particularly significant if the bolt material is not aluminum.

4-12.2 RIVETED JOINTS

Material strength compatibility between the rivet and structural members to be joined is very important—even more so than for bolted connections because bearing area compensation techniques are not possible. From an efficiency basis, the rivets and structural parts should have about equal strength properties[73,122]. However, it is often necessary to use somewhat softer rivets for rivet driving considerations. A hard rivet-soft plate material combination leads to excessive distortion and low bearing strength in the latter. The opposite combination obviously makes the rivet "the weaker link in the chain" and is not employed except in very light load applications—a situation not often encountered in military vehicles.

Hole clearance practice for rivets is a function of the riveting method used. Cold rivets are easily driven and require only small clearances; increased clearance is needed for hot riveting to speed insertion. Ref. 124 presents recommended hole sizes for these two riveting methods.

Durable riveted joints require that a number of design and production details be observed. Some of these details are illustrated in Fig. 4-72 by way of successful and unsuccessful results.

When a flush surface is required and the joint involves a thin sheet riveted to a much thicker member, as is common in automotive work, countersunk head rivets are often used. The procedure involves dimpling the thin sheet member to receive the rivet head and providing a mating countersink in the thick member. When the rivet is driven, the countersunk head rivet head forces the thin sheet into the countersunk hole provided in the thicker member, as shown in Fig. 4-73. The shear strength of this type of joint is substantially increased over a joint using conventional rivets[123]. Use of this technique is limited to the maximum thickness of sheet metal that can be dimpled easily—normally 0.064 to 0.091 in. (Ref. 124). Ease of dimpling is a function of the rivet diameter and the physical characteristics of the sheet material. The countersunk head rivet in a different application is shown in Fig. 4-72.

It is worthwhile to list some general rules of

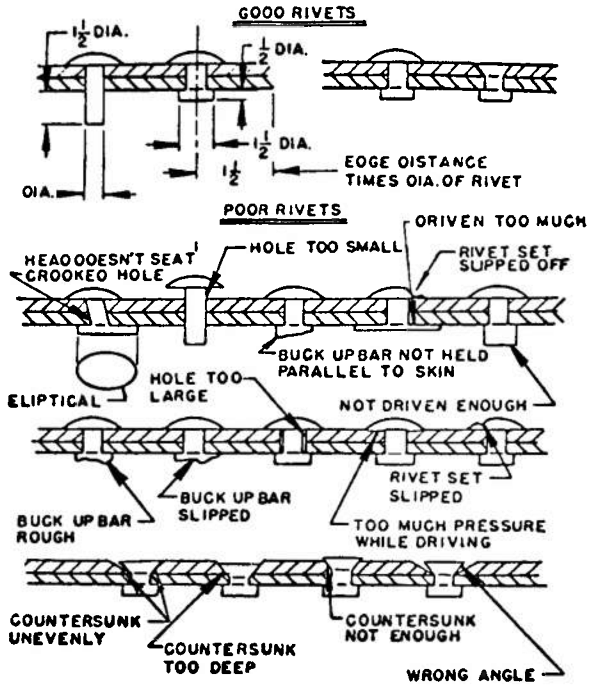


Figure 4-72. Successful and Unsuccessful Rivet Heads[125]

recommended riveted joint design[122,124,126].

a. Relations between dimensions of rivet and riveted parts

$$d \leqslant 3t$$
$$d \geqslant t'$$
$$d \leqslant 0.3w$$

where

d = rivet diameter

t = thickness of thinnest part to be joined (sheet, plate, rolled shape)

t' = thickness of thickest member to be joined

w = width of the leg of an angle.

b. Minimum spacing between centers of rivets in a row

$S_{(min)} = 3D$

Dimension $S_{(min)}$ is the same for threaded fasteners and is normally governed by the size of tool needed to drive or install the fastener.

c. Maximum spacing between rivets is usually set by the stability (buckling) limit of the sheet or plate. In instances where strength is not the controlling factor, a reasonable maximum spacing limit, such as $24t$ (t = sheet or plate thickness), is still set.

d. Edge distance (i.e. distance from edge of a

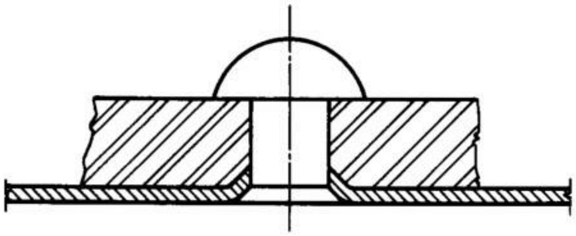


Figure 4-73. Flush Riveting With a Countersunk-head Rivet

part to be riveted to the center of the nearest rivet)

$S' \geqslant 2d$

Rivet arrangement and spacing are also influenced by the watertightness requirement of a particular joint. Watertightness of the joint will obviously have direct relation to rivet spacing. Riveted joints have many marine applications and the land vehicle designer with responsibility in the amphibious operations field can gain substantial information from the techniques of marine structural engineering. Oceangoing vessel construction uses the general rule of about four rivet diameters as the limiting spacing for watertightness in joint[127]. These data are in agreement with Refs. 122 and 126 which give other useful relations for pressure-tight riveted joints using caulked edges:

Spacing $3D \leqslant S \leqslant 4d$ or $S \leqslant 10t$

Edge distance $S' \leqslant 1.5d$ or $S' \leqslant 4t$

where

D = diameter of head

d = rivet diameter

t = sheet or plate thickness

See also Ref. 67 for additional riveting data.

Very little experimental work dealing with riveted joint watertightness has been published[113,114]. A significant exception, according to Ref. 127, is an investigation of aluminum alloy riveted joint behavior performed for the British Ship Research Association[128]. The objectives of the investigation were the parameters that affect joint strength, efficiency, and maximum allowable rivet spacing consistent with watertightness. As an indication of load intensity on the test joints, water pressure corresponding to a 15-ft head of water was used in the investigations. Excerpts of this work and its limitations are found in Ref. 127.

4-12.3 WELDED JOINTS

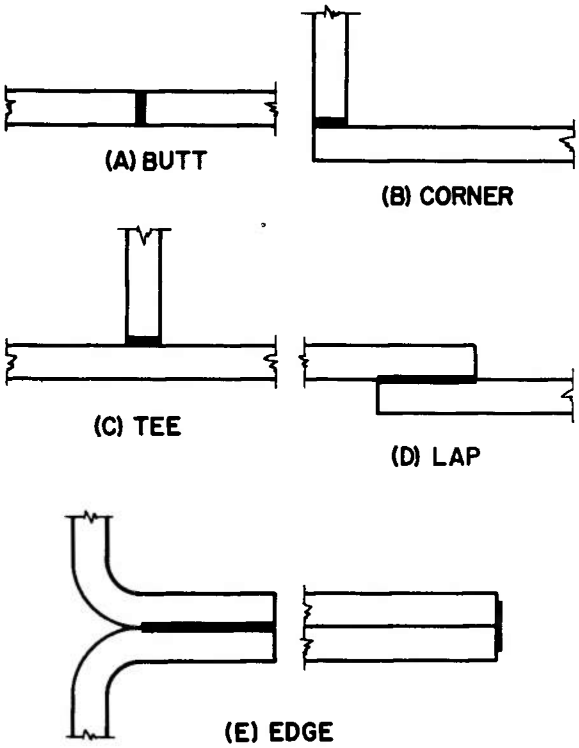
The master chart of welding processes, which appears in such sources as Refs. 129 to 131, illustrates the great scope of welding as a means of joint fabrication. For automotive construction, the processes primarily employed are limited to resistance, shielded arc, and gas welding techniques[130]. Each of these three welding methods comprises a family of specific processes. Shielded arc and gas welding are also often referred to under the single term of "fusion" welding. The particular welding method to be used depends on many factors of which vehicle type, joint material, joint accessibility, assembly sequence, speed of production, quantity of production, and economics are very important considerations.

Spot welding and seam welding, both members of the resistance welding family, find application in the structures of tactical military vehicles. The former process is the most common for general use, while the latter is ideal where greater sealing is needed—e.g., in roof, wall, and wheelhousing construction. In spot welding, either lap- or edge-type joints are used (par. 4-13 and Fig. 4-74); the spot welds may be single row, multiple row (par. 4-13) or can even include more complex patterns. As in riveting, edge distance and joint proportions are also important in spot welding. Minimum edge distance and recommended minimum-maximum spot weld diameters are functions of the sheet material thicknesses to be joined. This type of design data along with practical spot-welded joint design is given in Chapter 6 of Ref. 131.

Shielded arc and gas welding are processes more suitable to plate-type construction, which is characteristic of heavier vehicle applications or of those vehicles that require a very high degree of structural integrity (armored and nonarmored combat vehicles).

To increase welding production rates and remove the great variable of operator skill, semi- and fully automatic welding equipment is in general automotive use. The most sophisticated of these machines is limited to the higher volume production of nonmilitary automobiles. However, similar but less sophisticated machines—with simpler welding fixtures—are available for the production rates of military vehicle bodies and hulls.

A comparison of automatic continuous



BASIC JOINT TYPES ARE SHOWN WITHOUT REGARD TO WELD TYPE OR GROOVE SHAPE

Figure 4-74. Types of Welded Joints

welding with manual arc welding using intermittent welds has shown the former to give a more rigid construction and allow use of thinner material. Moreover, continuous bead arc and gas welding is also inherently suited for the fabrication of amphibious hulls in both the combat and tactical vehicle classes.

The joining of aluminum components requires use of special welding processes. The TIG (tungsten electrode, inert gas) and MIG (metal electrode, inert gas) welding processes have been highly developed, and welded aluminum fabrication is now accomplished exclusively by one of these two methods. Some advantages offered by the TIG and MIG processes are[126].

a. Welding can be performed in any position of the workpiece

b. Weld production rate exceeds that of nonshielded methods

c. Minimum of joint distortion can be realized because of the narrow heat-affected zone around the weld (usually <1 in.).

Weldability of the joint is another important design consideration. Data on the relative ease or difficulty of welding numerous materials or combinations of materials are readily available to the design engineer—commonly in tabular form[129,131-135]. An especially detailed coverage of weldability is found in Ref. 136, which, with Refs. 129, 130, 137, and 138, forms part of a very comprehensive series on welding.

4-13 TYPES OF JOINTS

The distinct types of joints used in automotive structures total only a small number. Of the three major methods of fastening joints discussed in the previous paragraph, the most numerous joint types are produced by welding. A larger number of joint types is possible with welding than with either bolting or riveting because welding is effective against more kinds of loading (par. 4-14).

Fig. 4-74(A) to (E) schematically illustrates the distinct types of welded joints. Use of all joint types for all welding processes is not implied. All the joints shown in Fig. 4-74 are used with fusion welding (shielded arc and gas), but only the lap and edge joints are applicable to spot welding. Welded joint types are considered in detail in Refs. 128, 131, 136, 139, 140, and 141. Refs. 140 and 141 are military publications that pertain to welded joint designs. Armored tank-type vehicles (or other structures that are to be subjected to ballistic attack) are covered by Ref. 140, while all other vehicles are treated by Ref. 141. In addition to coverage of joint types, these sources discuss the following important topics:

a. Compatibility of the numerous weld types with each joint type shown in Fig. 4-74

b. Joint edge preparation

c. Joint proportions, dimensions (member thicknesses, gaps, groove shapes and angles), and successful design details.

Only basic joint types are illustrated in Fig. 4-74, and no attempt is made to show the many variations that are possible by combining weld types and groove shapes.

The lap joint is the primary type for both bolted and riveted connections. It can be called the most basic joint type since it finds application in all joining processes and techniques. For bolted connections, the lap type is the sole applicable joint. Actually, the first of the edge joint configurations shown in Fig.

4-74(E) is also widely used with bolted joints for certain applications. However, in such cases it is known as a bolted flange joint, which in reality is a kind of lap joint. The most important joint type for riveted connections is also the lap joint. Two varieties of the lap joint exist: (1) the configuration given essentially in Fig. 4-74(D) (except, of course, that rivets are used as the fasteners) is a lap joint that subjects the rivets to single shear loading; and (2) in certain other applications, primarily boiler and pressure vessel fabrication, three plates (two from the same side of the joint) may form the lap joint, in which case the rivets are placed in double shear loading.

The riveted butt joint is given a separate classification in structural literature; although its appearance, action, and load handling modes are very similar to the lap joint. In the riveted butt joint, two plates are butted together (much as in Fig. 4-74(A)) and then fixed together by one, or preferably two cover plates[142]. This joint type is used mainly in boiler work. It is expensive to produce and is inherently heavy—attributes that make it unsuitable for automotive construction[143]. The term "multiple riveted" (e.g., double-, triple-, quadruple-riveted) used with lap and butt joints denotes the number of rows of rivets used in the fastener array.

Because they introduce weakened sections in the form of stress concentrators or heat-softened areas, both welded and riveted joints have less strength than the unjoined parent material. However, proper welding techniques and care make it possible to attain joint strength very nearly equal that of the parent material. Specifically in aluminum weldments, the optimum strength efficiency can be reached when the welded material is in its annealed condition[122,131]. In the case of heat-treated or cold-worked joints, a zone of metal on either side of the weld will have less strength than the parent metal due to the initial weld-heating. It has been found that butt welds are the most effective from the standpoint of weight, appearance, and joint life under repeated loading.

Substantial variations in the basic lap and edge joints of Fig. 4-74 are open to the vehicle designer for lighter weight sheet metal components.

In sheet metal construction, advantage should be taken of the formability of the material to produce strong, inexpensive joints. Figs. 4-75 and 4-76 show details of preferred sheet metal corner construction and practices to be avoided[67]. The flush-welded corner joint (Fig. 4-75) is preferred over the lap joints in cases where corrosion is prevalent or in cases where a plate is to be laid directly over the joint. The butt corner joint for thin section members (Fig. 4-76) is more costly than the lapped corner and has low strength. For cost reduction and increased strength, the spot-welded lap joint is preferred. The small corner opening that remains will require additional sealing if the corner forms a portion of an exterior structure.

4-14 LOAD TYPES AND MAGNITUDES ON JOINTS

4-14.1 GENERAL DISCUSSION

The direct reactions between fasteners and the structural parts united by them give rise to several types of loads in mechanical joints. The action of bolts and rivets on the joint components (sheets, plates, structural shapes) produces (a) bearing (compressive) loads and (b) tear-type (tensile) loads. On the other hand, joint components react on the fasteners to give (a) shearing loads and (b) crushing loads similar to the bearing loads developed in the plates or sheets.

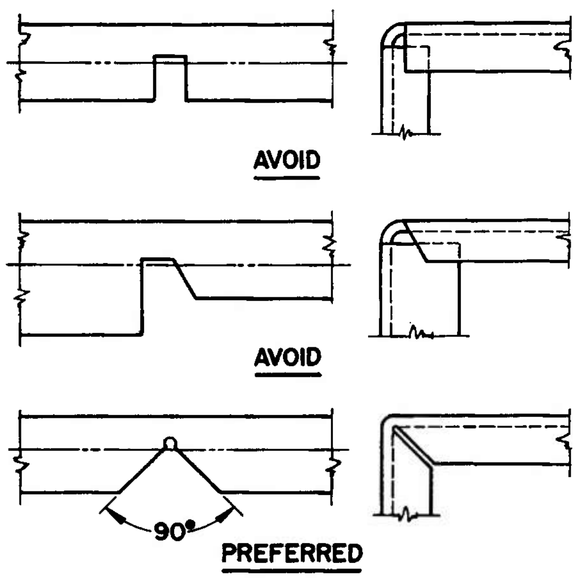


Figure 4-75. Sheet Metal Corner Construction for Welded Joints[109]

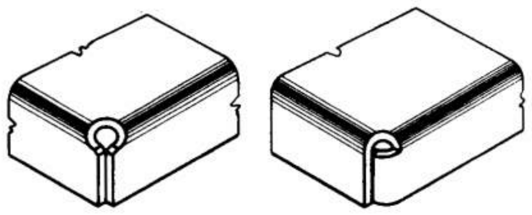


Figure 4-76. Joint Construction, Forming Round Corners[109]

In addition, tensile or separating forces can occur at the joint faying surfaces from the natural flexing action of structural members under load. Rivets have very little capacity to resist external tensile loads. In hot-rivet installation, considerable residual tensile stresses are present after cooling; thus, even small additional tension can cause early failure.

By contrast, bolted connections have substantial resistance to tension. However, a separating force between members works directly against the clamping force of the bolt, i.e., minimum clamping force remains at maximum applied tensile load. This underlines the importance of proper bolt preloading. Under dynamic loading, the effect on the bolts is a periodic elongation and relaxation that eventually leads to joint failure.

Eccentrically applied loads are very common in most joint configurations. The resultant bending moments give rise to still another tensile load (and stress) component in the fastener.

4-14.2 BEARING LOADS UNDER FASTENERS

Bearing stress S_b in the joint material (or in the fastener) is found by the relation

$$S_b = \frac{F_b}{A_b} = \frac{F_b}{dt}, \text{ lb/in.}^2 \qquad (4\text{-}85)$$

where

F_b = lateral fastener (or bearing) load, lb
A_b = projected bearing area, in.^{2}
d = fastener diameter, in.
t = total thickness of the joint, in.

Note that projected bearing areas of fastener and plate are taken to be the same quantity, even though a small difference may exist between the fastener and hole diameters.

For tear-type stresses S in sheets, plates, and similar parts, the following elementary equation is used for calculations

$$S = \frac{F}{A_n}, \text{ lb/in.}^2 \qquad (4\text{-}86)$$

where

F = tensile load on the part, lb
A_n = net cross-sectional area of the part (taken through a line of fastener holes), in.^{2}

Tear-type failures can be eliminated, or at least greatly minimized, by the use of proper edge distances, hole-to-hole dimensions, and observation of other pertinent joint proportions outlined in par. 4-12.

4-14.3 SHEAR LOADING IN FASTENERS

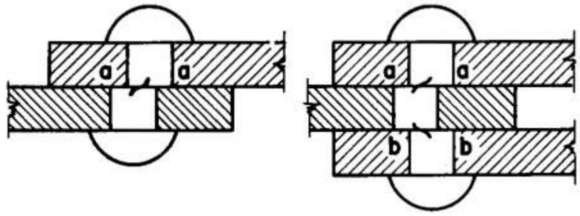
Shear loading on bolts (and more commonly on rivets) is classified as single shear or double shear according to the number of transverse load reversals along the length of the bolt or rivet. The difference between the two cases is illustrated in Fig. 4-77(A) and (B), respectively, for a rivet application carried to failure. The shear stress S_S in a single fastener loaded in single shear can be calculated with sufficient accuracy by the relation

$$S_S = \frac{F_S}{A} = \frac{4F_S}{\pi d^2}, \text{ lb/in.}^2 \qquad (4\text{-}87)$$

where

F_S = shearing force on the fastener, lb
A = cross-sectional area of the fastener, in.^{2}
d = fastener diameter, in.

In Eq. 4-87, cross-sectional area is based on nominal diameter of the fastener, even for a hot riveting application where the rivet expands to fill up the hole. When the joint is made up of multiple fasteners—fasteners loaded in double shear, or combinations of these—increase the value of A in Eq. 4-87 to include the total



(A) (B)

Figure 4-77. Shear Loading on Rivets[124]

fastener area subjected to shear loading.

Stresses S attributed to bending of fasteners can be examined by use of the standard beam equation

$$S = \frac{M}{Z} = \frac{Ft}{2Z}, \text{ lb/in.}^2 \qquad (4\text{-}88)$$

where

M = bending moment on the fastener, in.-lb

Z = fastener section modulus, in.^{3}

F = shear force or lateral fastener load, lb

t = grip length of fastener (total joint thickness), in.

Eq. 4-88 is not often used because its strict application demands a uniform load distribution over the fastener cross section. This requirement is rarely met in practice and, in addition, the loading is accompanied by local deformations of members. Nevertheless, Eq. 4-88 is useful in finding rough approximations of stresses due to bending of the fasteners.

In most instances it is the shear strength of the rivets (or bolts) and the bearing strength of the plate (or sheet) that governs the design. Since the lateral fastener load F_b, and the shearing force on the fastener F_S (Eqs. 4-85 and 4-87, respectively), and equal action and reaction forces; an interesting relationship of joint strength can be based on them

$$F_b = F_S = S_b dt = \frac{S_S \pi d^2}{4}, \text{ lb} \qquad (4\text{-}89)$$

or

$$d = \frac{4t}{\pi}\left(\frac{S_b}{S_S}\right) = 1.27t\left(\frac{S_b}{S_S}\right), \text{ in.} \qquad (4\text{-}90)$$

Eq. 4-90 describes a useful relation between fastener size and plate thickness in a well-designed joint based upon the actual stresses in the respective members. Since these stresses are not usually known in advance, the relation can be made more useful by introducing factors of safety f_b and f_s such that

$$f_b = \frac{S_b'}{S_b}\ ;\ f_s = \frac{S_S'}{S_S} \qquad (4\text{-}91)$$

where the primed values represent yield or allowable material stresses and the unprimed values, actual stresses.

Then

$$d = 1.27t\left(\frac{S_b'}{f_b}\right)\left(\frac{f_s}{S_S'}\right), \text{ in.} \qquad (4\text{-}92)$$

But, in an ideal joint, failure of the fastener and plate occurs simultaneously, or

$$f_b = f_s$$

and

$$d = 1.27t\left(\frac{S_b'}{S_S'}\right), \text{ in.} \qquad (4\text{-}93)$$

For example, a rivet material has a yield strength in shear of 34,000 lb/in^2 and the joint panel material has a bearing yield strength of 70,000 lb/in^2; then,

$$d = 1.27t\left(\frac{70}{34}\right) = (2.61)t, \text{ in.} \qquad (4\text{-}94)$$

For a given plate thickness, Eq. 4-94 gives the most economical size of rivet to use. It should be noted, however, that the relations given by Eqs. 4-89 through 4-94 apply only to plates (or sheets) equal to or thicker than ~5/32 in. (Ref. 123). A similar relation, based on the criterion of rivet shear strength equal to sheet buckling strength, can be written for lighter gages of material. This relation might be of interest for optimum sizing of rivets in lightweight vehicle design.

In a particular design problem, the stress values computed by Eqs. 4-85 to 4-87 should be compared to the yield or allowable stresses found in numerous sources; e. g., Refs. 122, 124, 128, and 133. The ratios of such allowable stresses to the computed stresses then serve as an estimate of safety factor for the joint. In a number of cases, joint strength is given in terms of allowable loads directly; for these instances, the simple ratio of allowable to computed load gives the safety factor.

The use of allowable strength values in bolted

aluminum joints requires caution. As mentioned in par. 4-12, full bolt strength can be used only if precise attention is given to bolt-to-hole geometry. When bolt holes in aluminum construction are more than a few thousandths oversize, a rule-of-thumb is to use only 2/3 of allowable bolt shear and bearing strengths for joint design[122]. Use of ultimate and yield bearing strengths in aluminum alloy parts are applicable only when proper distance is maintained between the center of a rivet and the edge of a member. Less than recommended edge distance (par. 4-12) requires reduction of allowable strength[122,126]. Ref. 133 recommends that actual material bearing strengths for nonstandard edge distances be determined by tests.

Strength considerations for countersunk rivets (par. 4-12) are similar to those for protruding head rivets, except that the effective bearing area has to be reduced. A common rule-of-thumb is to deduct one-half of the countersunk depth from the total plate thickness for the effective bearing thickness[122].

4-14.4 CLAMPING LOADS DUE TO BOLT TORQUE

The calculation of bolted joint strength is complicated by the largely uncertain state of friction between the faying surfaces. The required bolt tension or clamping force is known, but the actual clamping force obtained in an assembled joint is dependent on friction and, therefore, not sufficiently predictable. Where accurate measurements of bolt elongation in terms of tightening can be made, the resultant tension in a joint assembly can be readily determined. This method, however, is impractical in most cases including that of automotive structures and, as a result, empirical bolt load-to-torque relations have been established for convenience. The best known of the approximate bolt load-torque relations is

$$F = \frac{T}{Kd}\text{, lb} \tag{4-95}$$

where

F = bolt tension, lb
T = bolt torque, lb-in.
d = nominal bolt diameter, in.
K = torque coefficient, dimensionless $\cong 0.2$

Although the magnitude of the torque coefficient in Eq. 4-95 is only approximate, it does represent the average of measured values from a large number of tests performed on all sizes of bolts from ¼-in. to 1-in. diameter. Lists of ranges of torque coefficients appear in several references. For example, Ref. 144 shows these extreme values of the coefficient: 0.158 (-27 percent) to 0.267 (+33 percent); but average deviation is only 6 to 7 percent. Eq. 4-95 is general enough for use with all sizes of bolts and thread pitches. However, strict use of this equation is subject to the following restrictions[144]:

a. Semifinished hexagon nuts and bolts (width across flats ~1.5 times bolt diameter)

b. Free thread fits (excluding lock nuts)

c. Absence of bearing surface plating or lubrication

d. Bolt or nut must not be at a stress condition approaching failure

If written as functions of surface friction coefficients and bolt geometry, the expression for torque coefficient becomes more exact[144,145].

$$K = \mu_B \frac{r_B}{d} + \frac{r_T}{d}\left(\frac{\mu_T \sec\theta + \tan\phi}{1 - \mu_T \sec\theta \tan\phi}\right), \tag{4-96}$$

dimensionless

where

μ_B = friction coefficient at nut or bolt bearing surfaces
μ_T = friction coefficient at thread contact surfaces
r_B = friction force effective radius (at bearing surfaces), in.
r_T = friction force effective radius (at thread contact surfaces, in.
θ = thread half-angle, deg ($\sec\theta = 1.155$ for $\theta = 30°$, or 60° threads)
ϕ = thread helix angle, deg
d = nominal bolt diameter, in.

For threads normally found in fasteners, the term $(1 - \mu_T \sec\theta \tan\phi)$ is equal to unity ±½ percent (Ref. 144). Then, Eq. 4-96 can be written as

$$K = \mu_B \frac{r_B}{d} + \mu_T \frac{r_T}{d} \sec\theta + \frac{r_T}{d}\tan\phi, \tag{4-97}$$
$$= K_1 + K_2 + K_3$$

dimensionless

where

K_1 = portion of total torque effort wasted by friction at bearing surfaces of nut or bolt

K_2 = portion of total torque effort wasted by friction at thread contact surfaces

K_3 = portion of total torque effort that produces useful bolt tension.

Comparative values of K_1, K_2, and K_3 can be estimated if realistic physical dimensions are substituted into Eq. 4-97. Let:

$$\frac{r_B}{d} = 0.60$$

$$\frac{r_T}{d} = 0.40$$

$$\mu_B = \mu_T = 0.15$$

$$\sec\theta = 1.155 \text{ (60° threads)}$$

$$\tan\phi = 0.043 \text{ (2.5° helix angle)}$$

then

$$K = 0.090 + 0.069 + 0.0172 = 0.1762$$
$$= 51.1\% + 39.2\% + 9.7\% = 100\%$$

therefore,

$$K_1 \cong 50\% K$$

$$K_2 \cong 40\% K$$

$$K_3 \cong 10\% K$$

It is interesting to note that normally only about 10 percent of the total torque effort expended to assemble a threaded joint goes into actual production of clamping force. Of course, some variation is to be expected in the value of K_3 with such parameters as r_T, bolt size, and thread designation (coarse/fine). The variation of K_3 is direct with r_T and inverse with bolt size; magnitudes of K_3 are higher for coarse threads than for fine threads of the same bolt size. Ref. 144 also lists typical values of K_3 (as well as K, K_1, and K_2) found for a range of bolt sizes from 1/4-20 NC to 1-14 NF.

Torque coefficient is closely dependent on friction and is, therefore, subject to variations even under normally controlled conditions. Because of this, the experimental determination of torque coefficient is attractive when accurate, uniform, and reliable bolt tension is desired in joints. Precision torque-tension test instruments are available for this purpose[146], and are in use in the automotive as well as allied industries. An instrument of this type can determine reliable torque coefficients for particular joint designs. A test procedure is not practical on a production basis, but the true value of the instrument is that a few sample joints can be effectively used to simulate production joints. The simulation should reproduce actual joint conditions and should include at least the following provisions: use of correct joint components, surface hardnesses, lubrication (if any), and locking devices.

4-14.5 ECCENTRICALLY LOADED RIVETED OR BOLTED JOINTS

Eccentric loads on fasteners occur very frequently in automotive structures, in fact, whenever an appreciable moment arm exists between the fastener and the line of action of its load. A particularly important aspect of the topic concerns groups or arrays of fasteners under an eccentric load (Fig. 4-78).

When an eccentric load acts on an array of rivets (or bolts) in a joint, the resultant force on any one fastener is made up of two components—one due to direct load, and the other to a moment effect[134,139]. The direct load component is shared equally by all the fasteners, while the magnitude of the moment component on a fastener is proportional to its distance from the centroid of the fastener group (Fig. 4-78). Thus, the different fasteners of the array are not equally stressed.

The direct load component F_d on any one fastener is given by

$$F_d = \frac{P}{N}\text{, lb} \qquad (4\text{-}98)$$

where

P = total joint load, lb

N = total number of fasteners in the group.

The moment load component F_i on any fastener is

$$F_i = Cr_i\text{, lb} \qquad (4\text{-}99)$$

where

r_i = radial distance from the centroid CG of the total shearing area of the fastener group to any one fastener, in.

C = a proportionality constant to be found, lb/in.

i = designation of any fastener in the group = 1, 2, 3, ..., N.

The total external moment M on the joint is related to the individual moment loads on the fasteners by

$$\left.\begin{aligned} M &= Pe = \Sigma F_i r_i \\ &= C N_i r_i^2 \end{aligned}\right\} \text{ in.-lb} \qquad (4\text{-}100)$$

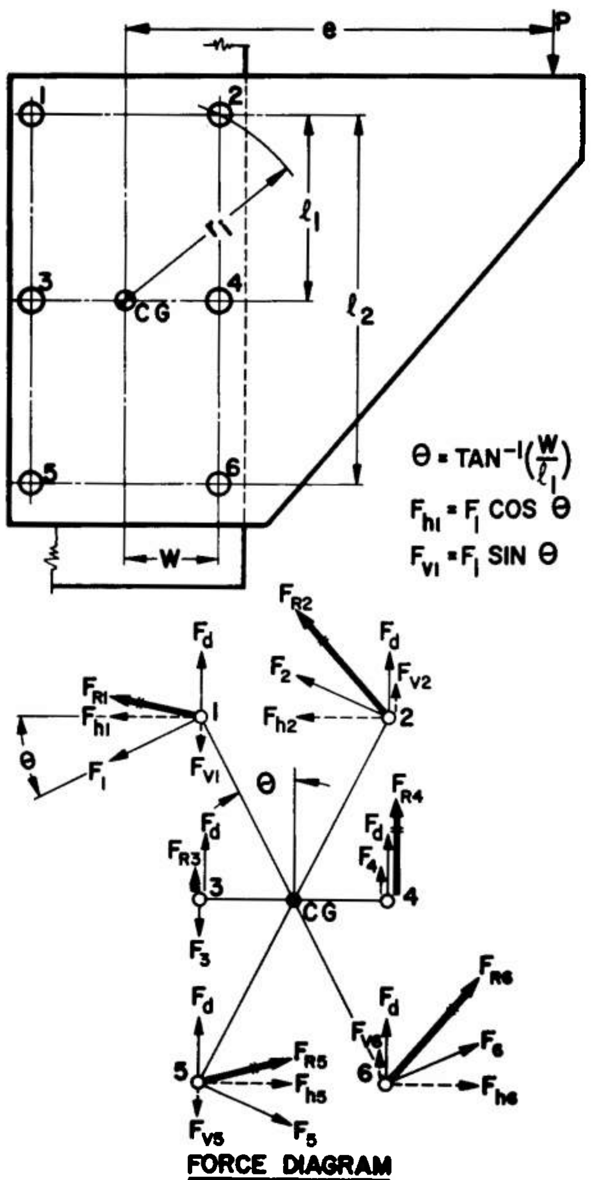


Figure 4-78. Eccentric Load on Fasteners

where

N_i = number of fasteners at a particular radius r_i

e = eccentricity of the load from the fastener group centroid, in.

M, P, F_i, r_i, and C are as previously defined.

Sample Problem

Given: A riveted joint of the configuration shown in Fig. 4-78.

$$\left.\begin{aligned} e &= 6 \\ \ell_1 &= 4 \\ \ell_2 &= 8 \\ w &= 2 \end{aligned}\right\} \text{in.}$$

P = 12,000 lb

N = 6

S_s' = 44,000 lb/in.^{2} (rivet shear yield stress)

S_b' = 70,000 lb/in.^{2} (plate bearing yield stress)

Find:

(a) The stress magnitude of the most heavily loaded rivet in the fastener group

(b) The ideal size of rivets for a total thickness of 1/4 in. in both plates of the joint and for the given material strengths

(c) Maximum rivet shearing stress

(d) Maximum plate bearing stress

(e) Factors of safety based on yield stresses

If the top plate in Fig. 4-78 is taken as a free body, and applying Eq. 4-98

$$F_d = \frac{P}{N} = \frac{12{,}000}{6} = 2{,}000 \text{ lb}$$

which is the direct load component. From Eq. 4-100

$$C = \frac{Pe}{N_i r_i^2} = \frac{12{,}000(6)}{4\,(\ell_1^2 + w^2) + 2w^2}$$

$$= \frac{12{,}000(6)}{4\,(4^2 + 2^2) + 2\,(2^2)} = 820 \text{ lb/in.}$$

Then, from Eq. 4-99, the moment load components are

$$F_{1,2,5,6} = Cr_{1,2,5,6} = C\sqrt{\ell_1^2 + w^2}$$

$$= 820\sqrt{4^2 + 2^2}$$

$$= 3670 \text{ lb}$$

$$F_{3,4} = Cr_{3,4} = Cw = 820(2) = 1640 \text{ lb}$$

Refer to Fig. 4-78,

$$\theta = \text{Tan}^{-1}\left(\frac{w}{\ell_1}\right) = \text{Tan}^{-1}\left(\frac{2}{4}\right) \cong 26°34'$$

Then,

$$\sin\theta = 0.4472; \cos\theta = 0.8944.$$

The moment load forces can now be resolved into horizontal and vertical components.

$$F_{h1,2,5,6} = F_{1,2,5,6}\cos\theta = 3670(0.8944) = 3280 \textit{ lb}$$

$$F_{v1,2,5,6} = F_{1,2,5,6}\sin\theta = 3670(0.4472) = 1640 \text{ lb}$$

Then the resultant forces on the various rivets are (Fig. 4-78)

$$F_{R1} = \sqrt{(F_d - F_{v1})^2 + F_{h1}^2}$$

$$= \sqrt{(2000 - 1640)^2 + 3280^2} = 3300 \text{ lb}$$

Similarly,

$$F_{R2} = \sqrt{(2000 + 1640)^2 + 3280^2} = 4900 \text{ lb}$$

$$F_{R3} = 2000 - 1640 = 360 \text{ lb}$$

$$F_{R4} = 2000 + 1640 = 3640 \text{ lb}$$

$$F_{R5} = F_{R1} \textit{ (magnitude only)} = 3300 \text{ lb}$$

$$F_{R6} = F_{R2} \textit{ (magnitude only)} = 4900 \text{ lb}$$

(a) Therefore, the most heavily loaded rivets are numbers 2 and 6.

From Eq. 4-93,

$$d = 1.27t\left(\frac{S_b'}{S_S'}\right) = 1.27(0.25)\left(\frac{70}{34}\right) = 0.505 \text{ in.}$$

(b) Therefore, use 1/2-in. rivets.

(c) From Eq. 4-87, maximum shearing strength in the rivets is

$$S_S = \frac{4F_S}{\pi d^2} = \frac{4(4900)}{\pi(0.5)^2} = 25{,}000 \text{ lb/in.}^2$$

(d) From Eq. 4-85, maximum bearing stress in the plate material is

$$S_b = \frac{F_b}{dt} = \frac{4900}{0.5(0.25)} = 39{,}200 \text{ lb/in.}^2$$

(e) From Eq. 4-91, factor of safety based on rivet shear is

$$f_s = \frac{S_S'}{S_S} = \frac{44{,}000}{25{,}000} = 1.76$$

Factor of safety based on plate bearing:

$$f_b = \frac{S_b'}{S_b} = \frac{70{,}000}{39{,}200} = 1.785$$

Therefore $f_s \cong f_b$.

This problem illustrates that load magnitudes vary significantly from one rivet to the next in a multiple fastener group. An analysis of this type can aid the designer in choosing an adequate number and size of fasteners as well as the proper arrangement of such fasteners for minimum moment loading.

4-14.6 ECCENTRICALLY LOADED WELDED JOINTS

Welded joints experience generally the same types of loading as riveted and bolted joints. Tensile stresses, whether direct or due to bending, and shearing stresses are the main concerns of the designer. Strength considerations must include both the weld metal and parent material.

Of the three general welding processes commonly used in automotive practices (par. 4-12), shielded-arc, arc-, and gas-welded joints are effective against all forms of stress because complete fusion takes place between joint members. However, the very popular spot welding technique (resistance welding family) is successful primarily against shear loads. Tensile load normally up to 45 percent of the allowable shear load is acceptable in well-produced spot welds[131]. In the presence of bending combined with shear, a variation of stress or "peeling" occurs across the spot weld. Spot-welded joints have a small but actual resistance against such peeling effects.

The equations in general use for design calculations of stresses in various types of fusion welded joints appear in Fig. 4-79[129]. In the case of a butt weld, the height dimension h shown in the equations does not include the reinforcement of the convex bead normally

found on this type of joint. Stress concentrations are common in welded joints—particularly in fusion welds where the transmitted forces often encounter drastic changes in structural cross section. Stress concentrations, particularly in welds, have been treated previously in par. 4-3.3.

In place of standard equations, the strength of spot-welded joints is determined with the aid of recommended design loads, empirically obtained for various sheet materials and joint configurations (Chapter 6, Ref. 131 and Ref. 133). For cases of special or peculiar configurations, load reliability can be obtained best by means of tests.

Other topics of interest concerning load types and magnitudes in welded joints are treated in Refs. 131, 133, and 134. These include:

a. Allowable tensile and shear stresses in gas- and arc-welded joints for various materials and alloys

b. Allowable parent material stresses near the welded zone

c. Allowable weld metal (electrode) strengths

d. Allowable tensile, shear, and peel loads for spot-welded joints as a function of sheet material thickness

e. Efficiency of spot-welded joints as functions of sheet material thickness and weld pitch (and distance between rows, in the case of multiple-row spot-weld patterns)

The coverage of spot-welding design is especially detailed in Ref. 133. Refs. 129, 130, and 136 to 138 comprise a comprehensive, periodically updated series on welding technology. Valuable design guidance for all types of welded joints can be found in these references.

Eccentric loads on welds produce a complex stress condition, but simplifying assumptions make possible a reasonable engineering analysis. The analysis is very similar to that presented earlier in this paragraph for eccentrically loaded joints with mechanical fasteners[134,139].

As in the mechanical joint, both a direct and a moment load component make up the resultant load in the eccentrically loaded welded joint. For the direct component, it is assumed that uniform stress distribution will occur across the weld area (basically indicative of static loads). The moment component is assumed to produce a stress at any point in the weld proportional to the distance of that point from the centroid of the weld pattern.

The direct stress component, Fig. 4-80, is

$$S_d = \frac{P}{A}\ , \text{lb/in.}^2 \tag{4-101}$$

where

P = total joint load, lb
A = total throat area of the weld, in.^{2}
= $0.707\ h\ell$ (for 45° fillet welds)
h = weld size, in.
ℓ = total weld length, in.

The moment or torque stress component at any point i in the weld pattern is

$$S_i = C' r_i,\ \text{lb/in.}^2 \tag{4-102}$$

where

r_i = radial distance from the centroid CG of the total weld shearing area to any point in the weld pattern, in.
C' = a proportionality constant, lb/in.^{3}

The total external moment or torque T on the entire weld group is obtained by integration of the moment force on an elemental area dA of the weld (Fig. 4-80).

$$T = Pe = \int S_i r_i dA, \text{lb-in.} \tag{4-103}$$

With the aid of Eq. 4-102,

$$T = Pe = C' \int r_i^2 dA = C'J = \left(\frac{S_i}{r_i}\right) J, \text{lb-in.}$$

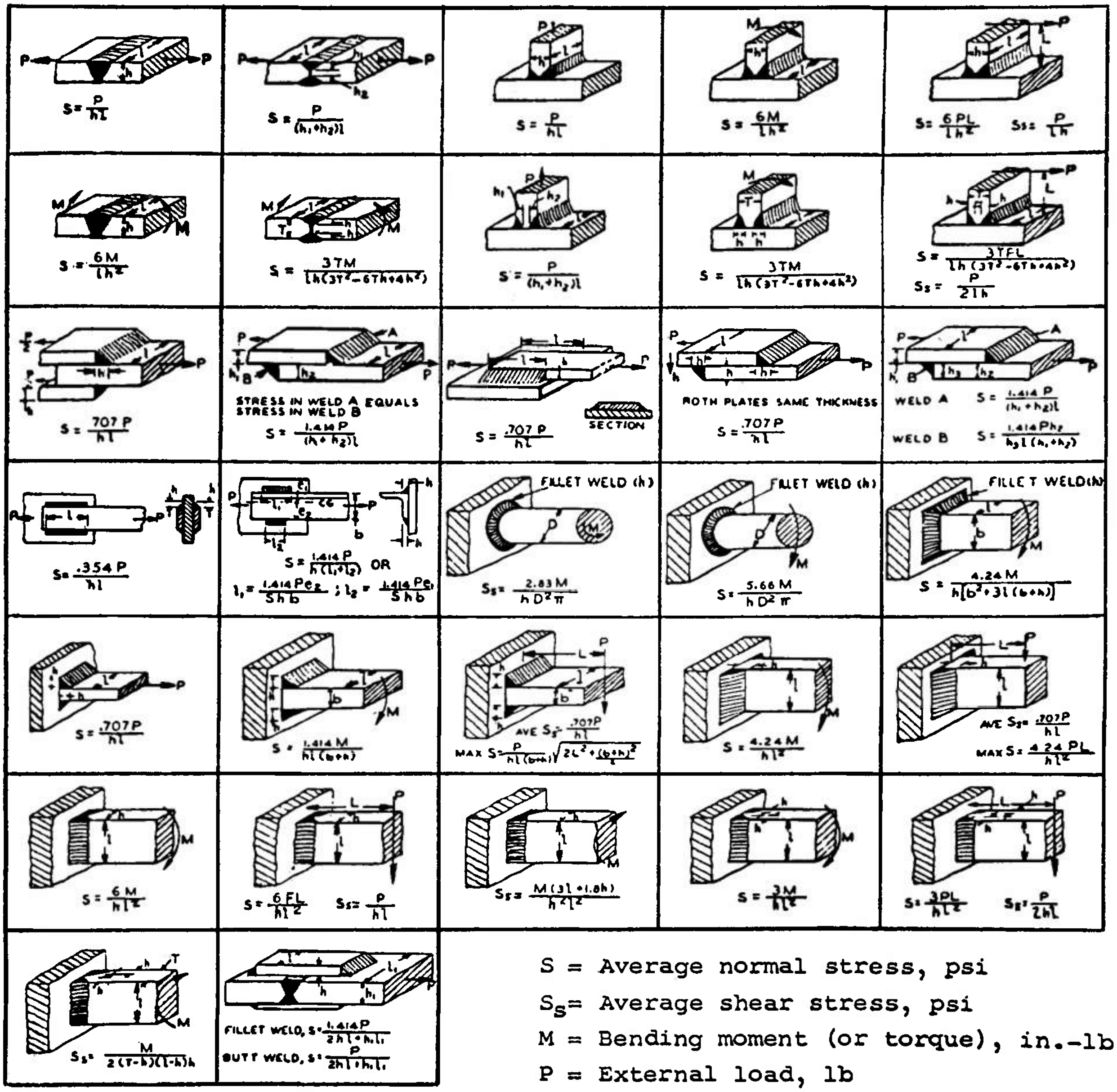
then, (4-104)

$$S_i = \frac{Tr_i}{J} = \frac{Per_i}{J}\ , \text{lb/in.}^2 \tag{4-105}$$

$$S_{i(max)} = \frac{Per_{(max)}}{J}\ , \text{lb/in.}^2 \tag{4-106}$$

where

e = eccentricity of the load from the centroid of the total weld shearing area, in.
$r_{(max)}$ = radial distance from weld shearing area centroid to the most distant point in the weld pattern, in.
dA = an elemental area of the weld, in.^{2}
J = $\int r_i^2 dA$ = by definition the polar moment of inertia of the weld shearing area about the centroid CG, in.^{4}



S = Average normal stress, psi
S_S = Average shear stress, psi
M = Bending moment (or torque), in.-lb
P = External load, lb
L = Linear distance, in.
ℓ = Weld length, in.
h = Weld size, in.

Figure 4-79. Weld Stress Formulas[129]

For calculation, the polar moment of inertia is written as

$$\left.\begin{aligned} J &= \sum\left(J_{oi} + A_i r_{oi}^2\right) \\ &= \sum\left(\frac{A_i \ell_i^2}{12} + A_i r_{oi}^2\right) \\ &= \sum\left[A_i\left(\frac{\ell_i^2}{12} + r_{oi}^2\right)\right] \end{aligned}\right\}, \text{ in.}^4 \qquad (4\text{-}107)$$

where

J_{oi} = polar moment of inertia of one weld segment in the group, about its own centroid, in.^{4}

r_{oi} = distance from the centroid of any weld segment to the centroid *CG* of the total weld pattern, in.

A_i = throat area of any weld segment in the pattern, in.^{2}
 = $0.707\, h\ell_i$ (for 45° fillet welds)

ℓ_i = length of any weld segment in the pattern, in.

The last two equations can be combined to give a single working relation for moment stress:

$$\left.\begin{aligned} S_i &= \frac{Per_i}{\sum\left[A_i\left(\frac{\ell_i^2}{12} + r_{oi}^2\right)\right]} \\ \text{or,} \\ S_{i(max)} &= \frac{Per_{(max)}}{\sum\left[A_i\left(\frac{\ell_i^2}{12} + r_{oi}^2\right)\right]} \end{aligned}\right\}, \text{ lb/in.}^2 \qquad (4\text{-}108)$$

The most heavily stressed portion of the weld pattern will be at the location where the vector sum of the direct and moment stress components is maximum. The similarity among Eqs. 4-101, 4-102, and 4-104 and Eqs. 4-98 to 4-100 should be noted.

Sample Problem

Given: A welded joint of the configuration shown in Fig. 4-80. Note that the joint proportions and loads have been kept the same as for the riveted joint sample problem given earlier in this paragraph.

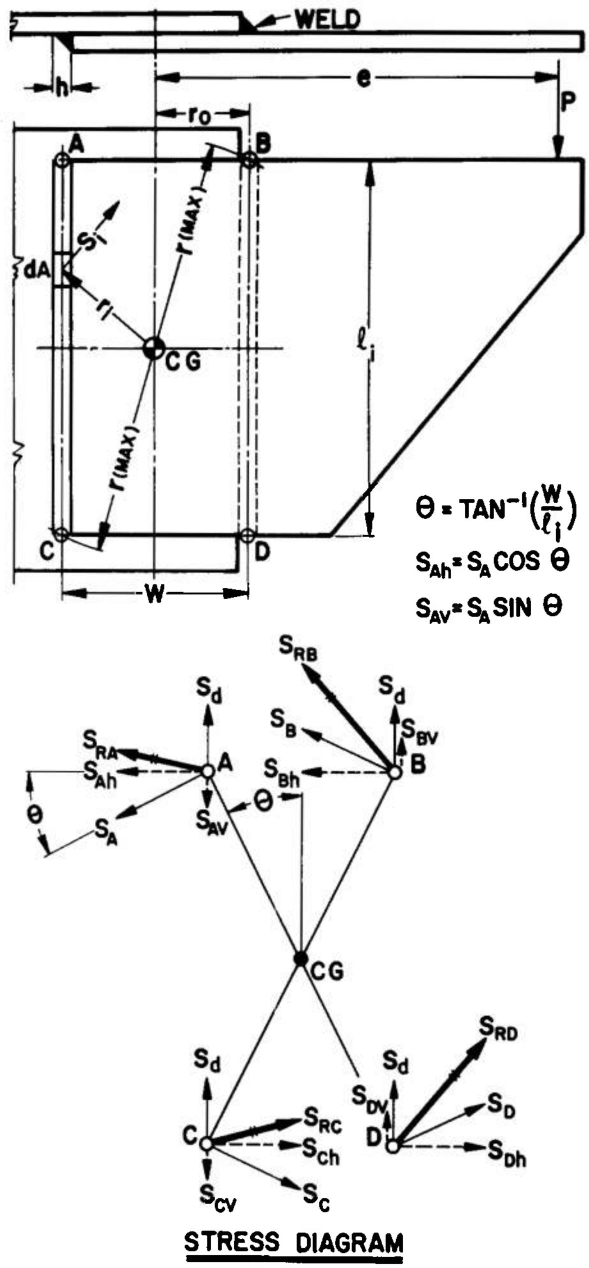


Figure 4-80. Eccentric Load on Welds

$$\left.\begin{aligned} e &= 6 \\ \ell_i &= 8 \\ w &= 4 \\ r_o &= 2 \end{aligned}\right\}, \text{ in.}$$

$r_{(max)} = \sqrt{\left(\frac{\ell_i}{2}\right)^2 + \left(\frac{w}{2}\right)} = \sqrt{20}$ in.

h = 1/4 in. (fillet weld size)

P = 12,000 lb

Find: The maximum stress in the welds. Refer to Fig. 4-80 and Eq. 4-101 and consider the top

plate as a free body.

$A = 0.707 \times 2h\ell_i = 0.707(2)(0.25)(8) = 2.828 \text{ in}^2$

The direct stress component has the same magnitude at any point in the weld pattern. Applying Eq. 4-101,

$$S_d = \frac{P}{A} = \frac{12,000}{2.828} = 4240 \text{ lb/in}^2$$

The moment stress component will be maximum at a point in the weld pattern that is farthest from the pattern centroid CG. For this particular problem, locations A, B, C, and D are all at the same maximum distance from the centroid of the weld pattern. From Eq. 4-108,

$$S_{i(max)} = S_{A,B,C,D} = \frac{Per_{(max)}}{\sum\left[A_i\left(\frac{\ell_i^2}{12} + r_{oi}^2\right)\right]}$$

$$= \frac{12,000(6)\sqrt{20}}{2\left\{1.414\left[\frac{(8)^2}{12} + (2)^2\right]\right\}}$$

$$= 12,200 \text{ lb/in}^2$$

The stress diagram in Fig. 4-80 (analogous to the force diagram of Fig. 4-78) shows that the maximum resultant stress will occur either at point B or D and will differ only in direction. Consider point B.

$$\theta = \text{Tan}^{-1}\left(\frac{w}{\ell_i}\right) = \text{Tan}^{-1}\left(\frac{4}{8}\right) \cong 26°45'$$

then,

$$\sin\theta = 0.4472;\ \cos\theta = 0.8944$$

$$S_{Bh} = S_B \cos\theta = 12,200(0.8944) = 10,900 \text{ lb/in}^2$$

$$S_{Bv} = S_B \sin\theta = 12,200(0.4472) = 5,450 \text{ lb/in}^2$$

The maximum stress in the weld at point B (or D) is the resultant S_{RB} of these two components of the torisonal stress S_B and the direct component S_d; thus

$$S_{RB} = \sqrt{(S_d + S_{Bv})^2 + S_{Bh}}$$

$$= \sqrt{(4240 + 5450)^2 + 10,900^2}$$

$$S_{RB} = 14,600 \text{ lb/in}^2$$

SECTION VI—TANK-TYPE BODIES*—CONSTRUCTION AND DESIGN

4-15 SPECIAL CONSIDERATIONS

4-15.1 CODES AND REGULATIONS GOVERNING DESIGN OF TANK BODIES

While tank-type bodies for military trucks and trailers are not required to adhere to the same ICC regulations that govern their civilian counterparts, these regulations can serve as a guide to general design practices and safety features that should be considered in the overall design. However when using civilian codes, it is important to remember the differences between the civilian and military vehicle environments. The military environment may allow a partial relaxation of civilian safety codes, since mission performance requirements may preclude the incorporation of all of the "nice-to-have" safety features required on a civilian vehicle. On the other hand, the cross-country capability requirement of military vehicles imposes structural loads that would soon render useless a vehicle designed to the allowable civilian vehicle stress level specifications. Also, the military design is far more flexible in the use of new materials than is the civilian design. For instance, in the transportation of Class B-type flammable liquids, the civilian vehicle designer[149] is limited to using tanks of ferrous metals or aluminum; whereas the designer of military tank-type bodies can use plastics or any other material that offer a design advantage without degrading the safety of the vehicle. The codes and regulations presented here should be used as a design guide, only; and it is expected that the designer will use, amend, or discard those portions of the regulations that will best allow an optimum overall military vehicle design.

*The terms "tank," "tank-type," "tank-like" applied to vehicle bodies in this section are descriptive of specialized bodies designed for the transport of liquid, gaseous, or granular bulk cargoes and are not to be confused with combat vehicles generally known as tanks.

The applicable regulations that govern the design of tank-type bodies are found in the *Code of Federal Regulations, Title 49—Transportation*[147] (CFR). In particular, parts 71-90 and 190-197 can be applied. Parts 190-197 are applicable to the design of any motor vehicle and are concerned with general safety regulations. Parts 71-79 apply to the transportation of explosives and other dangerous articles (parts 80-90 are reserved for later use). Of particular interest are Parts 78.321 (Specification MC300), 78.323 (Specification MC302), 78.324 (Specification MC303), 78.325 (Specification MC304), 78.326 (Specification MC305), 78.330 (Specification MC310), 78.331 (Specification MC311), and 78.337 (Specification MC331) which cover the design and construction of cargo tanks. The listed regulations cover cargo tanks fabricated of aluminum or ferrous metals and include the regulations governing the designs of cargo tanks used for the transportation of corrosive liquids and compressed gases.

Each CFR also cites additional codes and regulations that apply to the particular type of tank covered by the regulation. Ferrous metal structures must, generally, comply with the *ASME Boiler and Pressure Vessel Code*, Sections VIII and IX (the latest applicable revisions are cited in each CFR Part). Aluminum structures are required to have structural properties equal to those of the ferrous metals. Materials are generally required to meet ASME or ASTM requirements. The ASME code also governs the design of vents, valves, joints, metal thicknesses, welding, inspection, and testing. Other codes, such as those of the Chlorine Institute or of the Compressed Gas Association, are cited where applicable in each CFR Part.

4-15.2 BULKHEADS FOR TANK BODIES

Bulkheads are required (by the CFR) in every liquid cargo carrier that can be expected to transport its cargo at less than 80 percent capacity. The bulkheads are required to prevent the contents of a partially empty liquid cargo carrier from causing vehicle instability by sloshing during vehicle motion. The CFR requires that every cargo tank that has a total capacity in excess of 3,000 gallons shall be divided by bulkheads into compartments, none of which shall exceed 2,500 gallons. Military fuel tank trucks, such as the XM438E1 Goer, however, have single-compartment tanks up to 5,000-gallon capacity. These trucks, or comparable capacity trailers, have transverse or longitudinal baffles to reduce the liquid sloshing effect. The XM438E1, for instance, has two corrugated aluminum baffles spaced equidistantly inside the tank.

The CFR requires each baffle to have an area at least as great as 80 percent of the cross-sectional area of the tank. Bulkheads are required to have adequate strength to sustain a horizontal force equal to the weight of the contents of the tank between them and any adjacent bulkhead or tankhead. Fig. 4-81 illustrates two typical bulkheads designs that can be used and one bulkhead configuration that is *not* recommended.

4-15.3 LOADING, UNLOADING, AND VENTING

The general requirements for loading and unloading explosives and other dangerous cargoes are governed by the CFR, Title 49, Chapter 1, Subpart B, Parts 77.834 through 77.841. The CFR parts cited do not list on-vehicle components required but are useful as general background, including static electricity considerations. The CFR does not state specific requirements for loading and unloading components, except for the requirement that pumps and compressors be equipped with suitable pressure-activated by-pass valves that will permit flow from discharge to suction or to the tank, unless they are of the centrifugal type.

Present military tank-type vehicles have bottom loading and unloading capabilities. The same pump is used for both operations. Pumps for water tank vehicles are capable of generating a flow rate in the order of 60 gallons per minute. Pumps on fuel tank vehicles are capable of generating about 250 gallons per minute. In flammable liquid and gas carriers it is mandatory that all pumps and other electrical equipment that are in operation during loading or unloading be explosion-proof.

In addition to the bottom loading capabilities of tank-type vehicles, manholes are provided on the top of the structure for emergency top loading and for access to the tank interior for cleaning. A manhole is provided for each tank compartment.

Vents are mounted on tanks to prevent the

accumulation of gaseous vapors which may be generated during transportation and storage or to provide a relief of any excess pressure or partial vacuum which may be generated. For nonpressurized tanks, a simple inverted U-tube rising from the top of the tank or manhole can be used. Other vents are of the ball or diaphragm type. Pressurized tank-type bodies require special pressure relief valves which are activated before the overpressure reaches a value of 110 percent of the design pressure (see Part 78.337-9, Ref. 147).

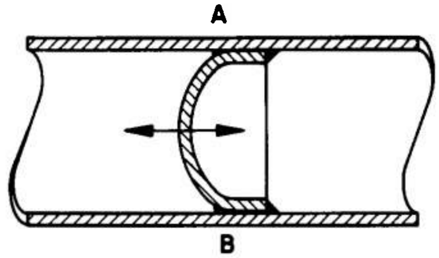
4-15.4 CLEANING

Periodically, tank-type trucks and trailers must be cleaned or purged to remove residue which builds up during use. To facilitate this operation, the designer must be aware of the methods of cleaning that are employed so that the cleaning operation can be accomplished in the most efficient manner. One method of tank cleaning requires an attachment that usually fits onto, or into, the large manhole of each tank compartment. Steam and chemicals, or simply chemicals alone, are then applied under pressure to the tank interior and the tank is washed clean. The apparatus that sprays the inside of the tank has a flow capacity in the order of 50 gallons per minute. Since standard military tank-type vehicles employ dispensing pumps that exceed this capacity, they can be used to discharge the expended cleaning fluid either into a tank for recirculation, into a disposal container, or into a disposal area. The use of the on-board pump allows the whole system to be cleaned as a unit. This type of cleaning, if used frequently, does a fairly efficient job as long as the residue has not become too thick or hard.

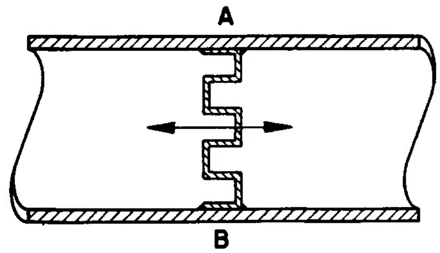
If the time between cleaning periods becomes too long, or if the residue cannot be adequately eliminated by the mechanically applied method, it becomes necessary for personnel to enter the tank and clean the inside by hand. For personnel to work effectively inside the tank, the atmosphere inside the tank must not be toxic, harmful to the occupant, nor be a potentially explosive mixture. To accomplish these requirements, the tank must be emptied and air must be circulated through the tank at a rate about twice the tank capacity every minute. To allow adequate circulation, an air outlet should be installed in the bottom of each tank compartment. In addition, tank drains should be installed to allow the solvent to drain from the vehicle. The same outlet can perform both functions.

4-15.5 TANK LININGS AND THERMAL INSULATION[147,148]

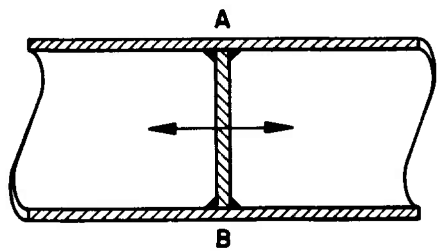
Tank linings serve two types of functions—(1) they prevent the contents of the tank from attacking the tank structural material, and (2) they provide thermal insulation for the tank contents. The CFR states that "The material used for lining cargo tanks shall be homogeneous, nonporous, imperforate when applied, not less elastic than the tank structure,



(A) DISHED BULKHEAD, MINIMIZES LOCAL BENDING AT A AND B. GOOD IN FATIGUE



(B) CORRUGATED BULKHEAD, LOCAL BENDING AT A AND B IS SPREAD OVER DEPTH OF CORRUGATION



(C) STRAIGHT BULKHEAD, HEAVY LOCAL BENDING AT A AND B. NOT RECOMMENDED

Figure 4-81. Typical Bulkheads

and substantially immune to attack by the commodities transported therein. It shall be of substantially uniform thickness, and it shall be directly bonded or attached by other equally satisfactory means. Joints and seams in the lining shall be made by fusing the material together or by other equally satisfactory means."

Not all tanks are lined. If the tank contents do not react with the tank structure, or if insulation is not required (such as in plastic or Fiberglas water tanks), a lining is not necessary.

Lining materials presently in use include phenolics, rubber, and plastics. Each type of material has its own characteristics, and usefulness. Most are used for protection of the tank structure against the action of corrosive contents. Some of the materials have an additional value as an insulator. Phenolics are relatively cheap to install, costing about half as much as rubber liners. They are applied by spraying the material on the inside of the tank and are oven-cured to obtain a hard glossy finish. The required liner thickness is obtained by applying several thin coats. While phenolic liners offer excellent corrosive resistant coatings, they are easily chipped which makes frequent inspections necessary. Small chips, however, are readily patched and cured locally.

Rubber liners are useful for lining a wide variety of tanks. They have good resilient properties as well as good corrosion and abrasion resistance. The rubber liners are installed by bonding sheets of rubber to the tank and vulcanizing the seams.

Plastics, such as polyvinyl chloride (PVC) or TFE-fluorocarbon, have been used as tank lining materials. One problem that occurs with plastic lining material is the difficulty in obtaining a satisfactory bond between the plastic and the tank. Generally, PVC is applied by bonding a rubber sheet to the tank and then bonding the PVC to the rubber. Seams are sealed by fastening plastic tape to the plastic laminate. This type of lining forms a very smooth surface which is easily cleaned.

To facilitate lining, the inside of the tank should be as smooth as possible with a minimum of sharp corners, depressions, and projections. Welds should be ground smooth and all weld spatter removed. Sharp corners should be ground off to achieve a smooth radius, and all fillets should be as generous as possible.

Properties of thermal insulating materials generally differ greatly from those of liner materials. Thermal insulating material has an open cellular structure which must be sealed when used in the inside of the tank. Conventional insulating techniques utilize an outer shell placed around the structural tank shell to provide a space which is then filled with insulation. Some development tank trucks, like the XM150E1 Water Tank Truck, utilize a sandwich-type tank construction consisting of two walls of glass-fiber-reinforced polyester resin with an insulating material bonded between the two walls. Besides effectively insulating the tank, this type of construction offers a significant weight saving over conventional metal construction.

4-16 LOADS AND LOAD DISTRIBUTIONS[147]

Tank-type bodies are subjected to a variety of loads which include hydrostatic internal pressure, bending, and shear loads. The loads are a result of the static forces generated by the cargo and the dynamic forces generated by the vehicular travel or transportation. The static and dynamic loads and load factors, which are applicable to any military vehicle, are presented in Section I of this chapter. Other load factors can be found in Ref. 147, Parts 78.315 through 78.337, which apply to the design of civilian tank-type vehicles. It will be found that, in most cases, the load factors required to satisfy the military environment will exceed those that satisfy the civilian environment. The CFR requirements are adequate for the design of some tank-type vehicles such as the M131A2 Tank Semitrailer which was designed to comply with ICC regulations. These vehicles, however, have only limited cross-country travel capabilities.

The maximum stresses in cargo tanks do not always occur when the tank is fully loaded with cargo. Deflections as well as stresses must be considered in the tank design analysis. In the past, tanks were supported on relatively large, heavy, rigid truck or trailer frames. As a result, the terrain shocks transmitted into the frame by the suspension system caused only minor frame deflections and virtually no tank deflections. Since it is highly desirable to design a minimum weight vehicle, the vehicle designer has

minimized the supporting frame structure and has often eliminated it entirely. The tank, then, must be designed so that the terrain shocks are carried by the tank skin as well as by any supporting framework. The amplitude and frequency of the terrain-induced shock transmitted by the suspension system into the vehicle structure is dependent, among other things, upon the weight of the vehicle (for a given suspension system). Therefore, a suspension system that is designed to produce only small deflections at low frequency in a heavily loaded vehicle can cause the empty vehicle to sustain local deflections that exceed the capacity of the structure because of the higher natural frequency of the unloaded suspension system[148].

Cargo tanks are attached to truck frames, to trailer frames, or directly to the fifth wheel kingpin support structure and the tandem wheel support structure. In addition, various items, such as the landing gear, piping, and auxiliary equipment are attached to the tank. Tanks generally fail from fatigue in areas subjected to high bending stresses. Since each point of tank or equipment attachment is a potential point of stress concentration, care should be taken to distribute the loading over as large a tank area as possible. Support pads should be generous, welds should be tapered to avoid abrupt section changes, and supports should be stiff enough to minimize the transmission of local deflections into the tank shell.

4-17 DESIGN CALCULATIONS[147,150-158]

4-17.1 SCOPE OF PROBLEM

The design calculations presented here for a tank-type cargo body are, by necessity, a simplification of an actual tank design. An actual design is an iterative process that encompasses input parameters and considerations from many interrelated vehicle design areas. The design problem presented here is limited to one series of calculations of the type which would normally serve as a "first cut" in the design process.

The tank body selected for the design analysis is a compressed-gas cargo tank designed to transport liquefied propane. The tank is considered to be a part of a semitrailer vehicle and to be mounted directly to the fifth wheel kingpin support structure and to the tandem wheel support structure with no external support frame. The requirements of the CFR Part 78.337 (MC331) will be met and the *ASME Boiler and Pressure Vessel Code* will be used as design criterion so that the reader will be somewhat aware of the regulations that govern the design of civilian cargo tank-type vehicles. The military vehicle designer will have to use applicable load factors to the design, and he is free to use a wider variety of materials; but, in general, the design procedure illustrated can be applied.

4-17.2 DESIGN PRESSURE

The design pressure is the maximum allowable working pressure of the tank and is dependent upon the cargo. Part 73.315 of Ref. 147 lists design pressure for various gases. For liquefied petroleum gas, specific gravity 0.509, the design pressure is equal to the vapor pressure of the gas at 115°F. Fig. 4-82 is a plot of the vapor pressure of commerical propane as a function of temperature. From this figure the design pressure of the tank is selected as 235 psig.

4-17.3 MATERIAL SELECTION

The tank is required to be of a seamless or welded steel construction[147]. The materials available for use on the tank are restricted to those that are permitted in Subsection C of Ref. 152 and to those described in Refs. 155 to 157. Refs. 155 to 157 are case interpretations of the boiler code which describe high-strength quenched and tempered steels that may also be used in the tank design. These steels have significantly higher strength characteristics than those in Subsection C of Ref. 152. In general, a lighter weight, greater payload vehicle can be designed by using the steels discussed in Refs. 155 to 157. For the example problem, the quenched and tempered steel described in Ref. 155 will be used. The composition and physical properties of this steel are given in Table 4-17.

Part 78.337-3(b) of Ref. 147 states that the vector summation of the stresses shall not exceed 25 percent of the minimum specified tensile strength of the metal. The maximum allowable stress of the steel is, therefore, 28,750 psi.

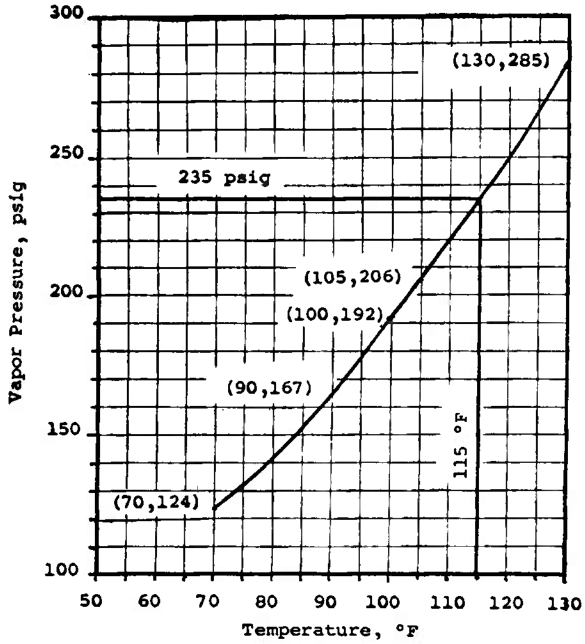


Figure 4-82. Vapor Pressure As a Function of Temperature, Commercial Propane[151]

4-17.4 FABRICATION

The method of tank fabrication shall, in general, agree with the requirements of Subsection B, Ref. 152. Welding procedures and tests shall be in accordance with Ref. 153. Joint fabrication requires that all butt joints be double welded and all longitudinal shell welds be located in the upper half of the tank.

All tanks constructed in accordance with Refs. 155 to 157 and all other tanks above 3,500-gallon capacity shall be provided with a manhole conforming to par. UG-46(g)(1) of Ref. 152 (Part 78.337-6, Ref. 147).

Part 78.337-13 of Ref 147 lists the requirements for tank supports and anchoring. For the system under consideration, with no supporting frame, the tank "shall be supported by external cradles subtending at least 120 degrees of the shell circumference". Maximum concentrated stresses which might be created at pads and cradles shall be calculated in accordance with Appendix G of Ref. 152.

4-17.5 INITIAL ASSUMPTIONS

Gross vehicle weight GVW = 50,000 lb

Tank is cylindrical with hemispherical ends.

TABLE 4-17 PROPERTIES OF QUENCHED AND TEMPERED STEEL[156]

Chemical Composition:	
Element	*%*
Carbon	0.15/0.200
Manganese	0.80/1.100
Phosphorous, max	0.035
Sulfur, max	0.040
Silicon	0.50/0.800
Chromium	0.50/0.800
Molybdenum	0.18/0.280
Zirconium	0.05/0.150

Tensile strength: 115,000 to 135,000 lb/in.^{2}

Yield strength: 100,000 lb/in.^{2}

Elongation in 2 in., min: 18%

External equipment weight:

Tandem assembly	=	4,900 lb
Landing gear	=	225 lb
Rubplate and kingpin structure	=	700 lb
Piping and auxiliary equipment	=	725 lb
Manhole and cover plate	=	150 lb
Total		6,700 lb

Sheet thickness (cylindrical portion) = 0.437 in.

Head thickness = 0.250 in.

Note: The minimum plate thickness is governed by paragraph UG-16(b) of Ref. 152, by Part 78.337-3 of Ref. 147, or by Refs. 155 to 157 depending on which is greater. For this problem, the minimum thickness is 1/4 in. and is governed by Ref. 156.

Outside diameter of tank = 7 ft

4-17.6 TANK SIZE CALCULATION

Weight of Tank and Propane W_{TO}:

$$W_{TO} = GVW - \text{Wt of external equipment}$$
$$= 50{,}000 - 6{,}700 = 43{,}300 \text{ lb}$$

Weight of Steel Heads W_{SH}:

Outside diameter of sphere = 7 ft × 12 in./ft = 84 in.

Inside diameter of sphere = 84 - 2(0.25)
= 83.5 in.
Volume of OD = 310,340 in.^{3}
Volume of ID = 304,831 in.^{3}
Volume of steel = 310,340 - 304,831
= 5,509 in.^{3}
Specific weight of steel = 0.283 lb/in.^{3}
Weight of steel heads = 5,509 X 0.283
= 1,559 lb

Weight of Propane in Heads W_{PH}:

Volume of propane in heads = 304,831 in.^{3}
Specific weight of propane (p. 7-33 of Ref. 155) = (0.509) (0.0361 lb/in.^{3} of H_2O)
= 0.018 lb/in.^{3}
Weight of propane in heads
= (304,830) (0.018) = 5,487 lb

Weight of Cylindrical Portion of Tank (Steel Plus Propane) W_{CT}:

$$W_{CT} = W_{TO} - W_{SH} - W_{PH} = 43{,}300 - 1{,}599 - 5{,}487 = 36{,}254 \text{ lb}$$

Weight of Cylinder per Foot of Length (Steel Plus Propane) W'_C:

Outside diameter of steel cylinder = 84 in.
Inside diameter of steel cylinder
= 84 - 2(0.437) = 83.126 in.
Cross-sectional area of cylinder wall

$$= \frac{\pi}{4}(84^2 - 83.126^2) = 114.852 \text{ in.}^2$$

Cross-sectional area of cylinder inside

$$= \frac{\pi}{4}(83.126)^2 = 5{,}425.93 \text{ in.}^2$$

Weight of steel cylinder per foot of length
= (0.283) (114.852) (12) = 390.037 lb/ft
Weight of propane in cylinder
= (0.018) (5,425.93) (12)
= 1,172.217 lb/ft

$$W'_C = 390.037 + 1{,}172.217 = 1{,}562.254 \approx 1{,}562 \text{ lb/ft}$$

Length of Cylindrical Portion of Tank L_C:

$$L_C = \frac{36{,}254 \text{ lb}}{1{,}562 \text{ lb/ft}} = 23.21 \text{ ft}$$

The dimensions of this tank semitrailer are illustrated in Fig. 4-83. The placement of the external equipment is arbitrarily located for this illustration.

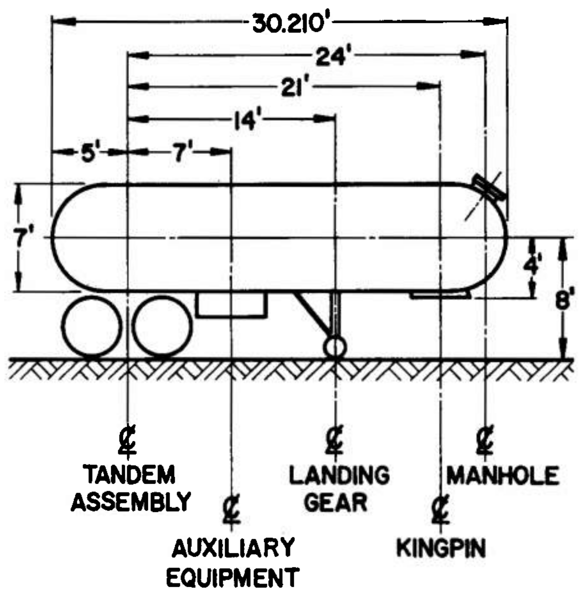


Figure 4-83. Semitrailer Dimensions

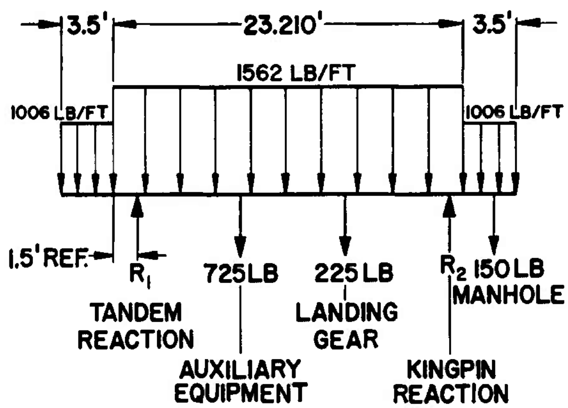


Figure 4-84. Tank Loading Diagram

4-17.7 TANK LOADING DIAGRAM

The tank loading diagram is illustrated in Fig. 4-84. For the purpose of simplification, the tank is assumed to be uniformly loaded at 1006 lb/ft at the hemispherical heads—5,487 lb of propane in heads plus 1,559 lb of steel in walls of heads, for a total weight of 7,046 lb, divided by 7 ft dia of sphere equals an assumed uniform loading of 1,006 lb/ft—and at 1,562 lb/ft at the cylindrical portion. The center of gravity of the hemisphere heads is assumed to be at their midpoint, i.e., 3.5/2 ft from their ends—because of the assumed

uniform loading. The tandem and kingpin reaction forces can be calculated by taking moments about the tandem reaction and then summing vertical forces.

Kingpin Tank Reaction R_2:

$$\Sigma M_{R_1} = (725)(7) + (225)(14) + (150)(24) - (1006)(3.5)\left(1.5 + \frac{3.5}{2}\right) + (1562)(23.21)\left(\frac{23.21}{2} - 1.5\right) + (1006)(3.5)\left(23.21 - 1.5 + \frac{3.5}{2}\right) - 21 R_2 = 0$$

$$R_2 = 449{,}332/21 = 21{,}397 \text{ lb}$$

Tandem Tank Reaction R_1:

$$R_1 = 725 + 225 + 150 + 43{,}300 - 21{,}397 = 23{,}003 \text{ lb}$$

4-17.8 SEMITRAILER LOADING DIAGRAM

Fig. 4-85 illustrates the trailer loading diagram. Included in the diagram are "G" loading effects specified in Part 78.337-3 of Ref. 147. A "2-G" loading is specified in all directions, and the side loading should be assumed as applied at the road surface. The section also defines the vertical 2-G load as "equivalent to three times the weight of the articles supported". This definition causes all of the loads in Fig. 4-84 to be multiplied by a factor of three when the tank stresses are calculated.

Load at Road at Center of Tandem Assembly F_1:

$$F_1 = R_1 + \text{wt of tandem assembly} = 23{,}003 + 4{,}900 = 27{,}903 \text{ lb}$$

Load at Kingpin F_2:

$$F_2 = R_2 \text{ wt of rubplate} = 21{,}397 + 700 = 22{,}097 \text{ lb}$$

4-17.9 TANK STRESS CALCULATIONS

The vehicle tank is subjected to stresses from the following loads:

a. Internal pressure
b. Method of tank support (Fig. 4-84)
c. Longitudinal acceleration (Fig. 4-85)
d. Lateral acceleration (Fig. 4-85)

The internal pressure loads result in a hoop stress $S_{T(H)}$ and a longitudinal stress $S_{T(L)}$.

Hoop Stress $S_{T(H)} = \dfrac{Pr}{t}$

where P = internal pressure = 235 lb/in.^{2}
r = internal radius of tank = 41.563 in.
t = thickness of tank = 0.437 in.

$$S_{T(H)} = \frac{(235)(41.563)}{0.437} = 22{,}351 \text{ lb/in.}^2$$

Longitudinal Stress $S_{T(L)} = \dfrac{S_{T(H)}}{2}$

$$= \frac{22{,}351}{2} = 11{,}176 \text{ lb/in.}^2$$

Fig. 4-86 is a shear diagram for the tank body under consideration when loaded as shown in Fig. 4-84 and incorporating the vertical load factor of three specified in Part 78.337-3 of Ref. 147 and mentioned in par. 4-17.8. A shear diagram is a graphic description of a loaded beam wherein the abscissa represents distances along the beam length and the ordinate values represent the magnitudes and directions of the vertical shear forces acting at the various sections along the beam. The bending moment at any station along the tank length is equal to the algebraic sum of the shaded areas in Fig. 4-86 lying between that station and the end of the tank; vertical shear values are considered as positive or negative according to whether they are directed upward or downward, respectively. In a structure of constant cross section loaded as

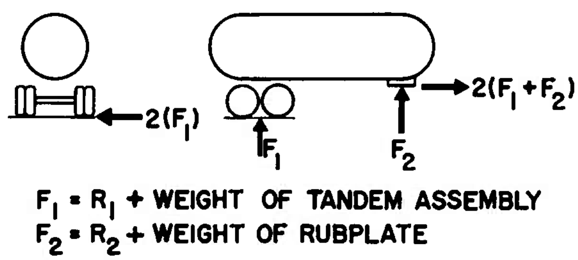


Figure 4-85. Semitrailer Loading

a beam, the maximum bending stress occurs in the section at which the bending moment is a maximum. This critical section is located at a point where the vertical shear is either equal to zero or else changes sign[1,2,5]. An examination of Fig. 4-86 reveals that, for the case being considered here, this critical section is point V_o at station 15.5 ft; and the maximum bending moment (and bending stress) due to tank loading occurs at this point.

Maximum Bending Moment Due to Tank Loading $M_{(max)}$:

$M_{(max)}$ occurs at point V_o in Fig. 4-86 and is equal to the algebraic sum of the shaded areas to the left of V_o; thus

$$M_{(max)} = (-10{,}563)\left(\frac{3.50}{2}\right)$$

$$+ (-10{,}573)\,(5.00 - 3.50)$$

$$+ (-17{,}592 + 10{,}563)\left(\frac{5.00 - 3.5}{2}\right)$$

$$+ (51{,}417 - 18{,}615)\left(\frac{12.00 - 5.00}{2}\right)$$

$$+ (18{,}615)\,(12.00 - 5.00)$$

$$+ (16{,}440)\left(\frac{15.50 - 12.00}{2}\right)$$

$$M_{(max)} = 234{,}281 \text{ ft-lb} = 2{,}811{,}372 \text{ in.-lb}$$

Tank Area Moment of Inertia I_T:

$$I_T = \frac{\pi(D^4 - d^4)}{64}$$

where

D = tank OD = 84 in.

d = tank ID = 83.125 in.

$$I_T = \frac{\pi(84^4 - 83.125^4)}{64} = 100{,}000 \text{ in.}^4$$

Cross-sectional Area of Tank Wall:

(Calculated Previously) A_T = 114.8 in.^{2}

Maximum Bending Stress from Tank Load $S_{TB(TL)}$:

$$S_{TB(TL)} = \frac{M_{(max)}D}{2I_T} = \frac{(2{,}811{,}372)(84)}{(2)(100{,}000)}$$

$$= 1{,}181 \text{ lb/in.}^2$$

Maximum Shear Stress from Tank Load $S_{S(TL)}$:

Maximum shear stress will occur at the point of maximum shear loading, which is at station 5.00 in Fig. 4-86. At this point the maximum shear load $V_{(max)}$ is 51,417 lb.

$$S_{S(TL)} = \frac{2V_{(max)}}{A_T} = \frac{(2)\,51{,}417}{114.8} = 896 \text{ lb/in.}^2$$

From Fig. 4-85 (and Part 78.337-3 of Ref. 147) it is noted that the tank has additional bending and tensile stresses due to the longitudinal acceleration force, which is equal to $2(F_1 + F_2)$—since a "2-G" loading is specified—and acts at a distance (moment arm) of 4 ft below the tank centerline (Fig. 4-83).

Acceleration-induced Tank Bending Moment M_A:

$$M_A = 2(F_1 + F_2)\,(4)$$

$$= 2(27{,}903 + 22{,}097)(4) = 400{,}000 \text{ ft-lb}$$

$$= 4{,}800{,}000 \text{ in.-lb}$$

Acceleration-induced Bending Stress $S_{TB(A)}$:

$$S_{TB(A)} = \frac{M_A D}{2I_T} = \frac{(4{,}800{,}000)(84)}{(2)(100{,}000)}$$

$$= 2{,}016 \text{ lb/in.}^2$$

Acceleration-induced Tensile Stress $S_{T(A)}$:

$$S_{T(A)} = \frac{2(F_1 + F_2)}{A_T} = \frac{2(27{,}903 + 22{,}097)}{114.8}$$

$$= 871 \text{ lb/in.}^2$$

Note: $S_{TB(A)}$ and $S_{T(A)}$ can be positive or negative depending upon whether the vehicle is accelerating or decelerating.

From Fig. 4-85 (and Part 78.337-3 of Ref. 147), the tank is subjected to an impact loading reaction, applied at the ground, which causes a torsional stress, a bending stress, and a shear stress to be exerted on the tank structure.

Torsional Moment M_T:

M_T = $(2F_1)$ (8) (from Figs. 4-83 and 4-85, and since center line of tank is 8 ft above ground)

= (2) (27,903) (8)

= 446,448 ft-lb = 5,357,376 in.-lb

Tank Area Polar Moment of Inertia J_T:

$$J_T = 2I_T = (2)\,(100{,}000) = 200{,}000 \text{ in.}^4$$

Tank Torsional Stress $S_{S(TSL)}$:

$$S_{S(TSL)} = \frac{M_T D}{2J_T} = \frac{(5{,}357{,}376)(84)}{(2)(200{,}000)}$$
$$= 1{,}125 \text{ lb/in.}^2$$

Side Loading Bending Moment M_{SL}:

The maximum tank stresses will occur at station 15.50 ft (at V_o of Fig. 4-86). M_{SL}, therefore, is calculated for this station.

$$M_{SL} = (2F_1)\,(15.50) = (2)\,(27{,}903)\,(15.50)$$
$$= 864{,}993 \text{ ft-lb} = 10{,}379{,}916 \text{ in.-lb}$$

Side Loading Bending Stress $S_{TB(SL)}$:

$$S_{TB(SL)} = \frac{M_{SL} D}{2I_T} = \frac{(10{,}379{,}916)(84)}{(2)(100{,}000)}$$
$$= 4{,}359 \text{ lb/in.}^2$$

Side Loading Shear Stress $S_{S(SL)}$:

$$S_{S(SL)} = (2)\,\frac{2F_1}{A_T} = \frac{(2)(2)(27{,}903)}{114.8}$$
$$= 972 \text{ lb/in.}^2$$

Table 4-18 summarizes the loads and tank stress levels.

Maximum Combined Stress $S_{(max)}$:

The maximum stress will occur at point V_o (station 15.50). In the following, S_x, S_y, and S_S represent longitudinal tensile stress, lateral tensile stress, and shear stress, respectively. At the top of the tank the following stresses are present:

$$S_x = S_{T(L)} - S_{TB(TL)} - S_{TB(A)} + S_{T(A)}$$
$$S_y = S_{T(H)}$$
$$S_S = S_{S(TSL)} + S_{S(SL)}$$

At the bottom of the tank the following stresses occur:

$$S_x = S_{T(L)} + S_{TB(TL)} + S_{TB(A)} + S_{T(A)}$$

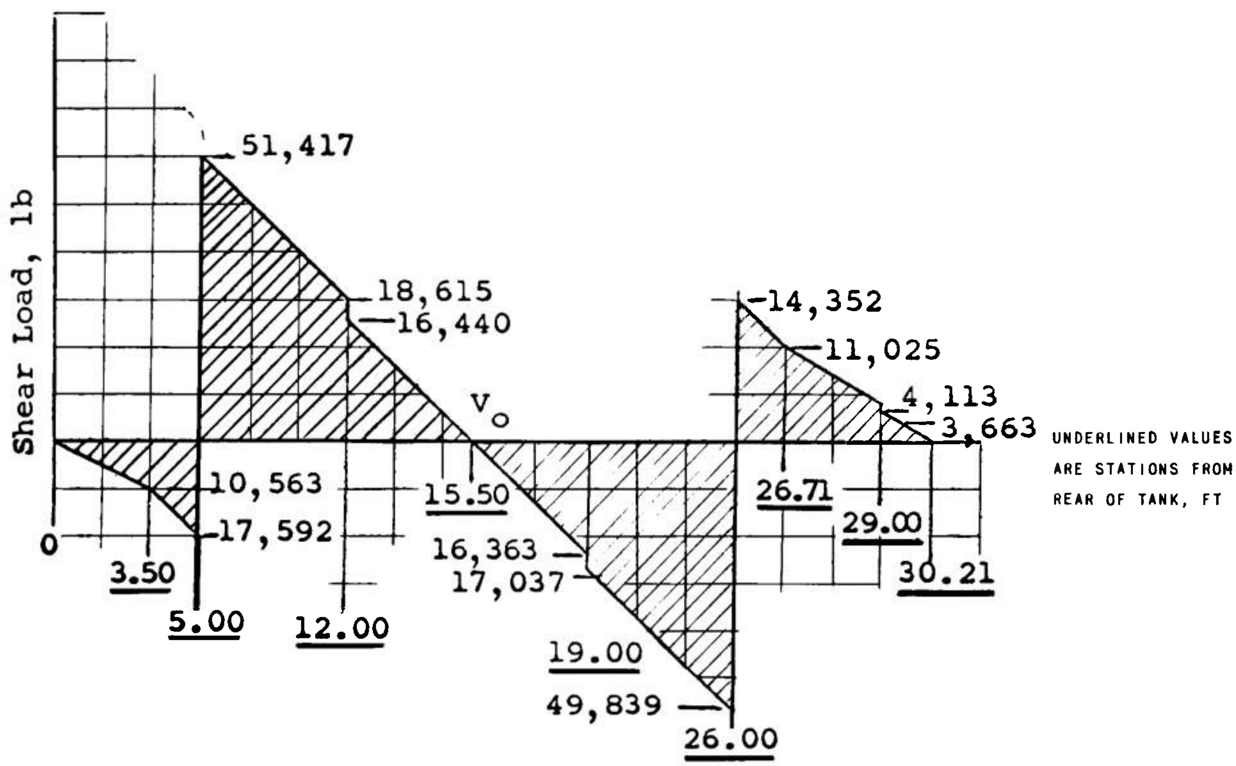


Figure 4-86. Tank Shear Diagram

$$S_y = S_{T(H)}$$

$$S_S = S_{S(TSL)} + S_{S(SL)}$$

In this example, the maximum stress will occur at the bottom of the tank.

$$S_x = 11{,}176 + 1{,}181 + 2{,}016 + 871 = 15{,}244 \text{ lb/in.}^2$$

$$S_y = 22{,}351 \text{ lb/in.}^2$$

$$S_S = 1{,}125 + 972 = 2{,}097 \text{ lb/in}^2$$

TABLE 4-18 SUMMARY OF LOADS AND STRESS LEVELS

Load Type	*Type of Stress*	*Stress Level, lb/in.²*
Pressure	Hoop Tensile $S_{T(H)}$ Longitudinal Tensile $S_{T(L)}$	22,351 11,176
Method of Tank Support	Bending (tension and compression) $S_{TB(TL)}$ Shear $S_{S(TL)}$	1,181 896
Longitudinal Acceleration	Bending (tension and compression) $S_{TB(A)}$ Tension or Compression $S_{T(A)}$	2,016 871
Lateral Impact (Side Load)	Torsional Shear $S_{S(TSL)}$ Bending (tension and compression) $S_{TB(SL)}$ Shear $S_{S(SL)}$	1,125 4,359 972

The maximum combined stress is obtained from the equation:

$$S_{(max)} = \frac{S_x + S_y}{2} + \left[\left(\frac{S_x - S_y}{2}\right)^2 + S_S^2\right]^{1/2}$$

$$= \frac{15{,}244 + 22{,}351}{2} + \left[\left(\frac{15{,}244 - 22{,}351}{2}\right)^2 + 2{,}097\right]^{1/2}$$

$$= 22{,}353 \text{ lb/in.}^2$$

The maximum stress generated is 22,353 lb/in.² The maximum stress allowable by the regulation is 28,750 lb/in.² Thus, the tank would be acceptable, although a little overdesigned. To complete the design, the designer must still consider the attaching structures such as the tandem assembly, tank support structure, and rubplate assembly. It is also to be noted that fatigue, which has not been considered in the example problem, must be considered in an actual design.

SECTION VII—AMPHIBIOUS OPERATIONS

An amphibious vehicle is one that is capable of operating satisfactorily on either land or water[158,165]. It may be tracked or wheeled and may be propelled by a screw, a jet, or by the action of its wheels or tracks. Amphibious vehicles are classed as floaters, swimmers, or true amphibians. Floaters are conventional land vehicles made buoyant by attached flotation devices. Swimmers are capable of traversing inland waters, such as lakes and rivers, under their own power. True amphibians are used in ship-to-shore operations and must, therefore, be capable of operating under moderate sea and surf conditions without capsizing or swamping.

The design of vehicles for amphibious operations requires a fundamental knowledge of the water environment plus the ability to relate the water effects to the stability and motion characteristics of the vehicle.

4-18 STABILITY[159,160]

Stability is the tendency of an object to return to its upright position when inclined away from that position. Stability may be considered as static stability or dynamic stability, depending upon the application. Static stability is a measure of the restoring moment at any particular angle of inclination. Dynamic stability is a measure of the amount of work that must be applied to achieve a particular angle of inclination. Dynamic stability, therefore, is the integration of the static stability-inclination angle curve.

Stability calculations require the determination of the vehicle's center of gravity, center of buoyancy, and its metacenter. The center of gravity is the point around which the vehicle mass is balanced. The center of buoyancy is

located at the center of gravity of the water displaced by the floating vehicle (displacement). The metacenter is the point of intersection of an originally vertical line (now inclined) through the center of gravity of a vehicle with a vertical line through the new center of buoyancy when the vehicle is heeled over (tilted).

Buoyancy is the term used to describe the upward force exerted by a fluid against a floating vehicle. This force is equal to the weight of the displaced volume of fluid. If the vehicle is to float, the weight of this displaced volume of fluid must equal, or exceed, the weight of the vehicle. This relationship, expressed in equation form, is:

$$F_B = V\rho \geqslant W_V \tag{4-109}$$

where

F_B = buoyancy force, lb

V = volume of displaced water, ft^3

W_V = weight of vehicle, lb

ρ = density of water, lb/ft^3

Since the location of the center of buoyancy is dependent upon the location of the center of gravity of the displaced water, any change in the displacement geometry, such as is caused by rolling and pitching of the vehicle, causes a displacement of the center of buoyancy. Thus, the center of buoyancy is not a fixed point, but rather a point about which the vehicle balances at any given instant. Since the center of buoyancy is a function of the rolling and pitching of the vehicle, the metacenter is a similar function. As the vehicle is tilted, the metacenter moves upward along the vehicle's vertical equilibrium axis.

Fig. 4-87 illustrates the relative static stability of three buoyant objects. Static stability requires that a vertical line passing through the center of buoyancy *(CB)* either be in line with the center of gravity *(CG)* of the vehicle, in which case the object is in equilibrium; or it must intersect the vertical equilibrium axis (metacenter) at some point above the center of gravity, in which case a righting moment is generated. When the metacenter *(MC)* falls below the center of gravity, an overturning moment is generated and the vehicle will capsize. These three stability relations are illustrated in Fig. 4-87(A), (B), and (C), respectively.

4-19 CENTERS OF PRESSURE

4-19.1 STATIC CENTER OF PRESSURE

The static center of pressure acts through the centroid of the volume of the water displaced. The integration of the pressure over the wetted area is equal to the buoyant force F_B.

4-19.2 DYNAMIC CENTER OF PRESSURE[164]

When an amphibian is at rest in the water, it has a particular angle of trim. If the vehicle moves, the angle of trim changes as the speed is increased. This action is familiar to anyone who has watched a speedboat's bow rise from the water as its speed is increased. The change in trim is caused by the generation of a dynamic lift force by the action of the water. In reality, the lift force is the summation of a dynamic pressure distribution over the wetted area. The point at which this force can be assumed to act is known as the dynamic center of pressure. Since the lift force (dynamic pressure) changes the vehicle's trim as the velocity increases, the wetted area, and hence the pressure distribution, changes. The changing pressure distribution alters both the magnitude and location of the center of pressure. Therefore, both the lift force and the center of pressure location are functions of vehicle velocity.

Each vehicle has a dynamic pressure distribution peculiar to itself. While crude estimates of this distribution can be made by using theoretical and empirical data, the normal procedure of obtaining the required dynamic data is through the use of a towing tank. Models of the proposed vehicle are towed at various speeds, drafts, and angels of trim. The drag, or resistance, and lift forces are measured. The center of pressure is obtained by measuring the moment generated about a known reference point. From the laws of similitude, the effects of velocity on the trim of the vehicle, the power requirements, and the maximum vehicle speed attainable (based on available power) can be estimated.

4-20 ROLLING[158,159,161]

Amphibious vehicles can roll in still water and when waves are present. In still water, rolling

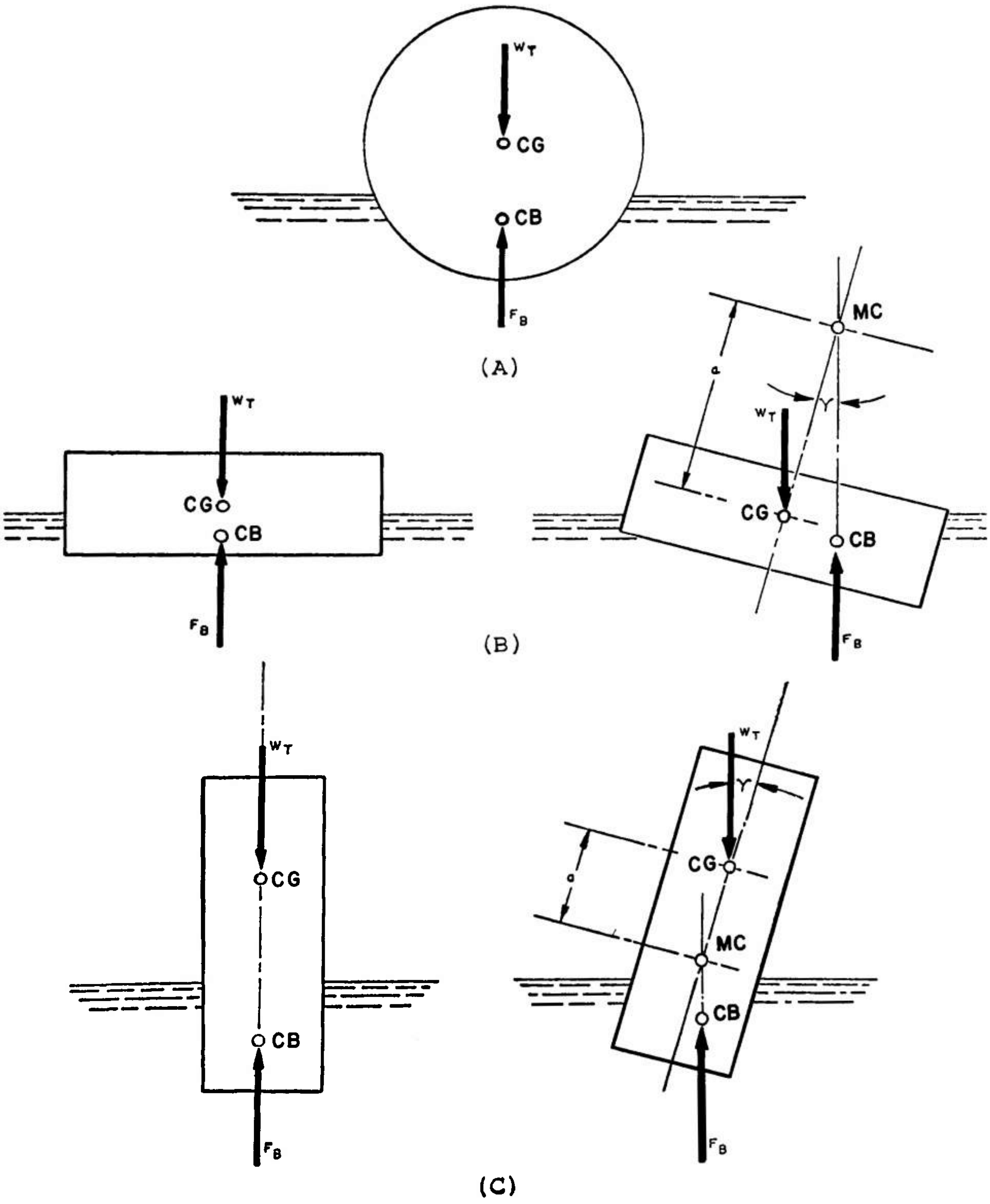


Figure 4-87. Stability of Buoyant Objects[158]

can be caused by a change in the vehicle's direction of movement, by a shifting of cargo, or by a shift of the center of buoyancy due to swells. When a vehicle rolls, a restoring moment acts to return the stable vehicle to an upright position. This restoring moment acts at each angular position, up to the equilibrium position, and thus gives the vehicle an angular rolling velocity. The rolling motion in still water is oscillatory, and the period of oscillation can be expressed by the equation:

$$T = 2\pi\sqrt{\frac{k^2}{ga}}\text{ , sec} \tag{4-110}$$

where

T = period of oscillation, sec
k = vehicle's radius of gyration, ft
g = gravitational constant, ft/sec^2
a = metacentric height (distance from center of gravity to metacenter), ft

It is readily evident that, for still water operation, a long period requires a long radius of gyration, a short metacentric height, or both. The general form of the equation of motion (assuming no rolling resistance) is

$$\gamma = \frac{\dot{\gamma}_o T}{2\pi} \sin \frac{2\pi t}{T} + \gamma_o \cos \frac{2\pi t}{T} \tag{4-111}$$

where

γ = angle of roll, rad
T = period of oscillation, sec
γ_o = angular displacement when t = 0, rad
$\dot{\gamma}_o$ = angular velocity when t = 0, rad/sec
t = time, sec

The maximum angle of swing $\gamma_{(max)}$ is found by

$$\gamma_{(max)} = \frac{\dot{\gamma}_o T}{2\pi} \quad \text{when } \dot{\gamma}_o \neq 0\text{, or by} \tag{4-112}$$

$$\gamma_{(max)} = \gamma_o \quad \text{when } \dot{\gamma}_o = 0 \tag{4-113}$$

For vehicles under the rolling influence of wave motion, the rolling resistance effect must be considered to supply meaningful information. Refs. 159 and 161 are useful in the derivation of the general equation of resisted rolling motion and are useful sources for background on many of the waveform assumptions and effects.

4-21 PLANING[161]

Planing is an activity normally associated with high-speed, shallow-draft boats and hydrofoils. As the speed of a shallow draft vessel is increased, the water supplies a dynamic lift force which is additive to the buoyancy force. If the dynamic center of pressure is forward of the vessel's center of gravity, the bow of the vessel is lifted. This action reduces the wetted area of the vessel, thus reducing the drag of the water, and allows the vessel to "plane" on the surface. The planing action permits relatively high speeds at moderate power requirements.

If the center of pressure is aft of the vessel's center of gravity, the dynamic lift force tends to depress the vessel's bow. Increasing the speed only tends to depress the bow further rather than to raise it. Many of the amphibious vehicles now in operation exhibit this bow-down tendency. Not only do these vehicles actively resist any planing tendencies, they are in danger of swamping at high speeds. One method used to offset this bow-down tendency is the installation of a "planing board" in the front of the vehicle. Then, as the speed is increased, the planing board generates a positive dynamic lift that counteracts the negative hull lift and allows the vehicle to trim out at a more level attitude.

Planing boards are also useful in operations in rough water. Under these conditions, the planing board reduces the amount of water that is splashed over the front of the vehicle. This eases the water removal requirements of the air intake, reduces the load on the bilge pumps, and aids in driver visibility.

Ref. 160 is useful in the design of planing characteristics of high-speed amphibious vehicles. The references at the end of Chapter 53 of Ref. 160 are particularly useful for many design applications.

4-22 LAUNCHING AND LANDING[162]

The launching and landing operations of amphibious vehicles generally require the successful crossing of surf, mud, marsh, soft sand, or banks. These conditions affect the

design of the hull in various ways. The same conditions that make an amphibious vehicle awkward and slow in the water make it a good surf-crossing vehicle. The amphibian normally has a low silhouette, limited buoyancy, and generally receives its water propulsion through the action of its tracks or wheels. In surf, the low silhouette limits the area against which the surf can act; the limited buoyancy causes a sluggish response to the heaving action of the surf and thus, inhibits vertical motion of the vehicle; and the tractive capability allows the vehicle to crawl on the ocean bottom when it is in the surf trough in relatively shallow water, which enables it to move forward between crests. Amphibians are, thus, noted for their ability to cross surf conditions that would swamp conventional landing craft. Because of their low freeboard, however, amphibians must have totally closed tops to keep from shipping water in high surf conditions.

Mud, marsh, and soft sand affect the shape of the hull only to the extent that the hull undersurface should be as smooth as possible to offer minimum motion resistance.

The negotiation of river banks is somewhat analogous to the launching of a ship. As the amphibian enters the water from a bank, it passes through a transition phase during which a portion is supported by the water through the buoyancy force. As the vehicle continues into the water, the buoyancy force rotates the vehicle into a more horizontal attitude until the vehicle reaches its trim attitude. At this time, all of the vehicle weight is supported by the water. If the bank is steep enough, the seaward end of the vehicle may become submerged if the righting moment generated by the buoyancy force is insufficient to rotate the vehicle sufficiently to keep the seaward end water-free. If the bank is too steep, water will enter open portions of the vehicle. Furthermore, if the vehicle enters the water at any appreciable speed, the seaward end can be driven deep below the surface of the water from the inertia effects of the moving vehicle mass. Normally, an amphibian is driven into the water, from a steep bank, slowly and cautiously to minimize these inertia effects.

Landing an amphibious vehicle is essentially the reverse of launching, and the same basic static condition considerations apply. If the vehicle is landing at an appreciable speed, however, the inertia effects of the vehicle mass must be considered. In the inertial landing mode, the seaward end of the vehicle tends to be driven underwater by the rotational inertia moment of the vehicle mass and the moment arm generated by the point of vehicle-bank contact. As in the launching sequence, the landing vehicle normally negotiates steep banks at slow speeds, which permits the assumption of static conditions in related calculations.

In certain situations where a vehicle approaches a relatively steep bank (or a bank that offers poor traction) at a very slow speed, the bank reaction force may not be great enough to supply sufficient traction for the vehicle to emerge from the water. In such cases, the vehicle has to be backed off a short distance and a landing at a higher approach speed attempted.

Fig. 4-88 illustrates the static force system acting on an amphibious vehicle during launching and landing operations. Initially, the vehicle floats level in the water such that its center of gravity *(CG)* and its center of buoyancy *(CB)* line up on a common vertical axis. In landing upon a steep bank, its front end makes contact at some initial point A' and then proceeds up the slope to some point A''. As the front end makes contact, the vehicle receives partial support S from the bank, which produces a couple Sa about the center of gravity. This couple tilts the vehicle to some angle β causing the stern to submerge deeper into the water. The resulting change in geometry of the displaced volume of water causes the center of buoyancy to shift toward the stern creating a couple $F_B b$. Since equilibrium conditions exist while the vehicle progresses up the slope, the couple due to the bank reaction must equal the couple due to buoyancy, or stated mathematically,

$$F_B b = Sa \tag{4-114}$$

where

F_B = buoyancy force, lb

b = horizontal distance from the vehicle center of gravity to the line of action of the buoyancy force, ft

S = vertical bank reaction force, lb

a = horizontal distance from the vehicle center of gravity to the line of action of the vertical bank reaction force, ft

Equilibrium conditions also dictate

$$F_B + S = W_V \tag{4-115}$$

where

W_V = vehicle weight, lb

The buoyancy F_B and its moment arm b are functions of the depth $Z_{A''}$ of the contact point and of the trim angle β. Similarily, the moment arm a is a function of the trim angle β. An analysis of the launching and landing conditions can best be accomplished by the use of graphical methods. The general procedure is described in the paragraphs which follow.

A likely point of contact on the vehicle is selected for each contact depth $Z_{A''}$ that is being considered, and the buoyancy F_B and its moment arm b are calculated for several values of β to cover the likely range of equilibrium $\beta \leqslant a$. Since W_V is known, S is readily calculated, using Eq. 4-115 for each value of W_V (and hence of β). The moments $F_B b$ and Sa are calculated for each value of β and are each plotted as functions of β. The intersection of these two curves represents the value of β at which the vehicle will be in equilibrium.

A scale drawing can now be made of the vehicle partially supported at the assumed point of contact A'' and displaying the trim angle of equilibrium that has been just determined. The amount of submergence of the outboard end (such as dimension C in Fig. 4-88) can now be studied. This procedure can be repeated for a number of assumed contact points and a number of slopes a to develop continuous curves of desired information as the vehicle moves up (or down) the slope during a landing or launching operation.

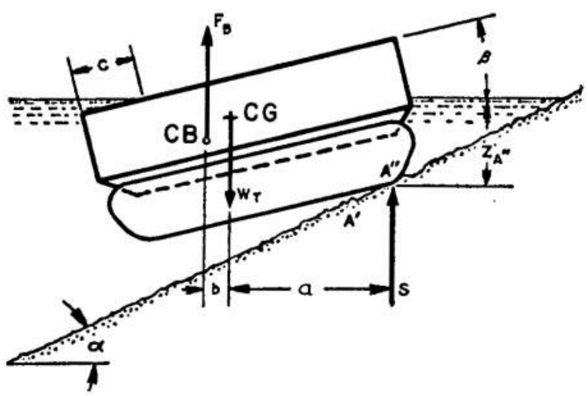


Figure 4-88. Force System Acting on an Amphibious Vehicle During Launching and Landing

4-23 DESIGN CALCULATIONS[159-164]

The computations involved in determining the hydrodynamic behavior of complex buoyant shapes are far beyond the scope of this handbook. In order to illustrate the procedures that are used, however, they were greatly simplified by considering the vehicle to be the simple rectangular prism shown in Fig. 4-89. This, then, is the basis for the following sample design calculations. Detailed discussions of the methods applied to the complex shapes of the real world are given in the references cited.

4-23.1 TRIM CALCULATIONS

The method used for calculating the natural trim of complex floating shapes is given in Ref. 159. For illustrative purposes, the simple vehicle shape shown in Fig. 4-89 is used here. Fig. 4-90 shows the general attitude that the vehicle will assume when afloat. The total displacement volume, shown crosshatched in Fig. 4-90 is divided horizontally into subvolumes 1 and 2 to facilitate the computations. In situations involving complex hull shapes, the total displacement volume is generally divided into vertical segments.

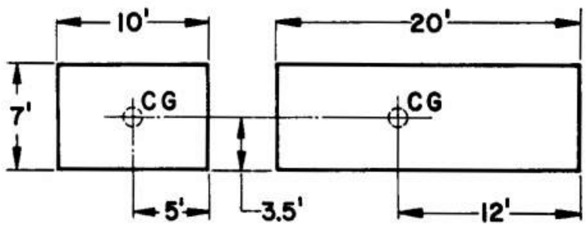


Figure 4-89. Illustrative Vehicle

Let:

L = length of vehicle, ft
H = height of vehicle, ft
B = breadth of vehicle, ft
W_V = weight of vehicle, lb
e = distance of vehicle center of gravity from bow, ft
V_1 = submerged volume 1
V_2 = submerged volume 2

F_{B_1} = buoyancy force from submerged volume 1, lb
F_{B_2} = buoyancy force from submerged volume 2, lb
F_B = total buoyancy force, lb
d = draft at bow, ft
β = angle of trim, deg
CG = center of gravity location of vehicle
CB_1 = center of buoyancy of volume 1
CB_2 = center of buoyancy of volume 2
ρ = density of water, lb/ft^3

Objective: To determine draft d and angle of trim β

$$V_1 = \frac{dBL}{\cos\beta} \tag{4-116}$$

$$F_{B_1} = \rho V_1 = \frac{\rho dBL}{\cos\beta} \tag{4-117}$$

$$V_2 = \frac{BL^2\tan\beta}{2} \tag{4-118}$$

$$F_{B_2} = \rho V_2 = \frac{\rho BL^2\tan\beta}{2} \tag{4-119}$$

Since the summation of moments about the vehicle center of gravity under equilibrium conditions equals zero (buoyancy forces act through CG of volumes 1 and 2),

$$F_{B_1}\left(\frac{e}{\cos\beta} - \frac{L}{2\cos\beta}\right) - F_{B_2}\left(\frac{2L}{3\cos\beta} - \frac{e}{\cos\beta}\right) = 0$$

$$F_{B1}\left(e - \frac{L}{2}\right) = F_{B_2}\left(\frac{2L}{3} - e\right) \tag{4-120}$$

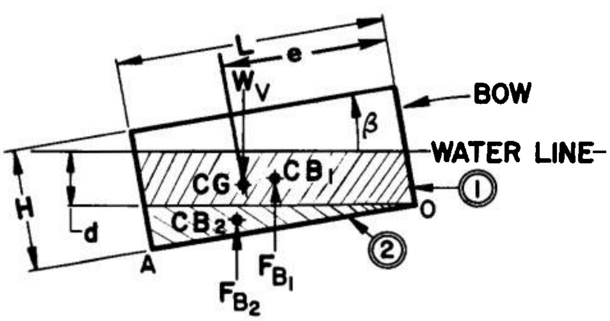


Figure 4-90. Vehicle in Water

Combining Eqs. 4-117, 4-119, and 4-120 and solving for d

$$d = \frac{L\sin\beta\,(2L - 3e)}{3(2e - L)} \tag{4-121}$$

Also, for equilibrium

$$F_{B_1} + F_{B_2} = F_B = W_V \tag{4-122}$$

Substituting Eqs. 4-117 and 4-119 into Eq. 4-122

$$\frac{\rho dBL}{\cos\beta} + \frac{\rho BL^2\tan\beta}{2} = W_V$$

from which

$$d = \left(\frac{W_V}{\rho BL} - \frac{L\tan\beta}{2}\right)\cos\beta \tag{4-123}$$

Equating Eqs. 4-121 and 4-123

$$\frac{L\sin\beta\,(2L - 3e)}{3(2 - L)} = \left(\frac{W_V}{\rho BL} - \frac{L\tan\beta}{2}\right)\cos\beta$$

from which

$$\tan\beta = \frac{\dfrac{W_V}{\rho BL}}{\dfrac{L(2L - 3e)}{3(2e - L)} + \dfrac{L}{2}} \tag{4-124}$$

Using Eqs. 4-121 and 4-124 along with the vehicle parameters, the trim angle and bow draft can be calculated.

Parameters:

W_V = 35,000 lb
L = 20 ft
B = 10 ft
e = 12 ft
ρ = 64 lb/ft^3

Trim Angle β:

$$\tan\beta = \frac{\dfrac{35{,}000}{(64)(10)(20)}}{\dfrac{20\,[(2)(20) - (3)(12)]}{3\,[(2)(12) - 20]} + \dfrac{20}{2}}$$

$\tan\beta$ = 0.164 [from Eq. 4-124]

β = 9.32°

Bow Draft d:

$$d = \frac{(0.162)(20)\,[(2)(20) - 3(12)]}{3[(2)(12) - 20]}$$

$$d = 1.08 \text{ ft}$$

Stern Draft: $= d + L \sin\beta$
$= 1.08 + (20)(0.162)$
$= 4.32$ ft

4-23.2 STABILITY CALCULATIONS

As the vehicle begins to roll, its center of buoyancy is displaced laterally from its position at zero roll. A similar displacement of the center of gravity also takes place. As the angle of roll increases, the buoyancy moment increases to some maximum value and then decreases as the roll angle continues to increase. The moment produced by the displaced center of gravity also increases as the roll angle increases up to a maximum value when the angle of roll is 90°. This gravitational moment can act with or against the buoyancy moment, depending upon whether the center of gravity is below or above the waterline. Fig. 4-91 illustrates the relative positions of the center of buoyancy, the center of gravity, and the center of roll for various angles of roll. It is apparent in the figure that the moments generated by the buoyancy force and the vehicle center of gravity change as the angle of roll is increased. In this figure, the restoring moment is equal to the moment generated by the buoyancy force minus the moment generated by the rotation of the center of gravity. If the center of gravity were below the waterline, the two moments would be additive.

The figure is somewhat simplified in that it shows the center of roll as a fixed point. In the rectangular prism illustrated, the center of roll remains fixed only between roll positions (A) and (B), Fig. 4-91. Between these positions, each incremental volume of the vehicle that enters the water is balanced by an equal volume that leaves the water, thus maintaining a constant displacement volume. In positions (C) and (D), Fig. 4-91, larger volume increments enter the water than leave, per incremental increase in roll angle. Thus in reality, the prism would float higher in the water, and the center of roll would be a function of the angle of roll.

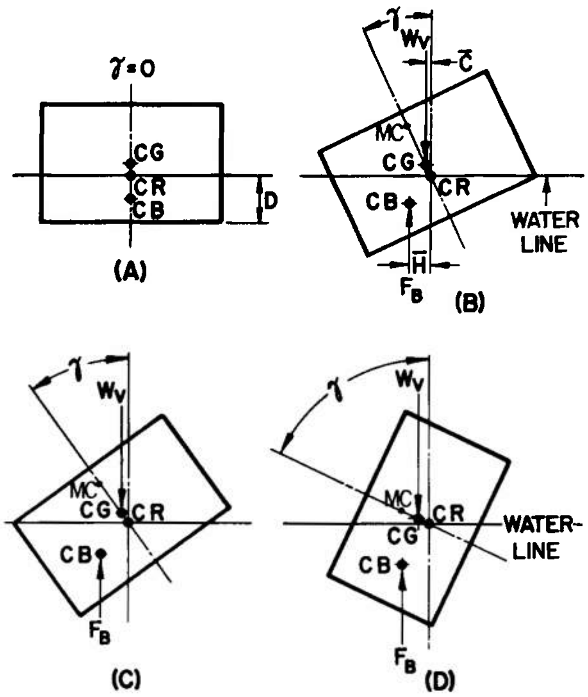


Figure 4-91. Relative Positions of Center of Gravity CG, Center of Buoyancy CB, and Center of Roll CR for Various Angles of Roll γ

The situation is further complicated by the angle of trim. It is evident, from Fig. 4-91, that the roll angle at which the condition shown at (B) occurs (the angle at which a portion of the hull bottom begins to leave the water) is a function of the draft *d*. When the vehicle has a trim angle, the draft at the bow and stern are different (see Fig. 4-90). If the vehicle illustrated is in a bow high attitude, the bow will reach a condition (B) roll while the stern is still at a roll angle between conditions (A) and (B). As the roll angle continues to increase, the bow buoyancy force tends to increase (since a greater vehicle volume is entering the water than is leaving), which causes the bow to rise relative to the stern. This action couples the roll angle to the trim angle and causes the trim angle to increase as the roll angle increases (after a certain initial roll angle). The vehicle can be shaped or loaded in a manner that will maintain the volumes entering and leaving the water constant as the roll angle increases. In this way, the trim angle can be made independent of the roll angle. In a similar manner, if the vehicle volume entering the water is less than that

leaving, the trim angle will tend to decrease as the roll angle is increased.

When making stability calculations where complex hull shapes are involved, the procedures given in Ref. 159 are recommended. In general, these require the following steps:

1. The trim angle is calculated as outlined in par. 4-23.1.
2. The vehicle is divided into a number of parallel transverse sections—the greater the number of sections selected the better will be the final accuracy, but the more laborious will be the procedure.
3. The draft, displacement volume, center of buoyancy, and buoyancy force of each section are determined for various roll angles up to the maximum being considered.
4. The resultant centers of buoyancy of the total displacement volume of the vehicle is determined from the preceding for each incremental angle of roll. This yields the moment arms $\bar{H}$ of the buoyancy moments (Fig. 4-91).
5. The positions of the vehicle center of gravity for each incremental angle of roll is determined. This yields the moment arms $\bar{C}$ of the gravity moment.
6. The tipping moments $W_V \bar{C}$ and the buoyancy moments $F_B \bar{H}$ are compared at each roll angle to determine the angle of instability—the angle at which the tipping moment is greater than the restoring moment.

4-23.3 ROLL CALCULATIONS

As an illustration of roll calculations, the following example is used based upon the simplified vehicle configurations shown in Figs. 4-89 and 4-90 and upon the equations for rolling motion given in par. 4-20. For the purpose of this example, the assumptions are made that the vehicle is rolling in calm water and that there is no resistance (damping) to the rolling motion.

The period of oscillation can be calculated by means of Eq. 4-110, which is repeated here for convenience.

$$T = 2\pi\sqrt{\frac{k^2}{ga}}$$

where

$$k^2 = \frac{I}{M} = \frac{I_g}{W_V}, \text{ft}^2 \tag{4-125}$$

M = vehicle mass = W_V/g, slug

W_V = vehicle weight, lb

I_g = moment of inertia of vehicle about longitudinal axis through the center of gravity, slug-ft²

The moment of inertia I of a rectangular prism about its roll axis such as is shown in Fig. 4-89 can be calculated from the following:

$$I = \frac{M}{12}(B^2 + H^2) \tag{4-126}$$

where B and H represent the vehicle breadth and height, respectively, in ft.

The metacentric height a in Eq. 4-110 can be calculated from the following:

$$a = \frac{\bar{H} - \bar{C}}{\sin\gamma} \tag{4-127}$$

where $\bar{H}$ and $\bar{C}$ are the lengths (ft) of the moment arms of the buoyancy and gravity moments, respectively, as shown in Fig. 4-91. In cases where the center of gravity is below the center of buoyancy, Eq. 4-127 must take the form

$$a = \frac{\bar{H} + \bar{C}}{\sin\gamma} \tag{4-127 alternate}$$

The parameters $\bar{H}$, $\bar{C}$, and a vary with the roll angle γ. Values were calculated for these parameters for roll angles varying from 0° to 60° and are shown in Tables 4-19. Since the value of a does not vary greatly over the range being investigated, an average value of 1.25 ft is used in the following computations.

Applying Eq. 4-126 and using a vehicle weight W_V of 35,000 lb as before (M = 35,000/32.2 = 1,087 slugs),

$$I = \frac{1{,}087}{12}(10^2 + 7^2) = 90.58\,(100 + 49)$$

$$= 13{,}496 \text{ slug-ft}^2$$

From Eq. 4-125,

$$k^2 = \frac{13{,}496}{1{,}087} = 12.42 \text{ ft}^2$$

The period of the oscillation is computed from Eq. 4-110:

$$T = 2\pi\sqrt{\frac{12.42}{(32.17)(1.25)}}$$

TABLE 4-19 METACENTRIC HEIGHT a AS A FUNCTION OF THE ROLL ANGLE γ

Roll Angle γ, deg	$\sin \gamma$	$\overline{H}$, ft	$\overline{C}$, ft	a, ft
0	0.000	0.00	0.00	. . .
5	0.087	0.13	0.04	1.02
10	0.174	0.28	0.08	1.14
15	0.259	0.43	0.12	1.19
20	0.342	0.59	0.16	1.25
30	0.500	0.92	0.23	1.37
40	0.643	1.23	0.30	1.44
50	0.766	1.43	0.36	1.40
60	0.866	1.42	0.41	1.17

$T = 3.48 \approx 3.5$ sec

The oscillatory rolling motion of the vehicle is described as a function of time by Eq. 4-111, which is repeated here for convenience.

$$\gamma = \frac{\dot{\gamma}_o T}{2\pi} \sin \frac{2\pi t}{T} + \gamma_o \cos \frac{2\pi t}{T}$$

In order to apply this equation, γ_o and $\dot{\gamma}_o$ must be known. For the purpose of this illustration, these were assumed as $\gamma_o = 0.087$ rad and $\dot{\gamma}_o = 1.0$ rad/sec. Eq. 4-111 was solved for various increments of time t from 0 to 35 sec. The results were first tabulated and then plotted. This plot of roll angle versus time is shown in Fig. 4-92.

The maximum roll angle $\gamma_{(max)}$, based upon the assumed values of γ_o and $\dot{\gamma}_o$, is determined by means of Eq. 4-112.

$$\gamma_{(max)} = \frac{\dot{\gamma}_o T}{2\pi}$$

$$\gamma_{(max)} = \frac{(1)(3.5)}{2\pi} = 0.557$$

$$\approx 32^\circ$$

4-24 HYDRODYNAMIC DRAG[159,160,163]

The drag, or resistance to motion, of an amphibious vehicle consists of air drag and water drag. When the vehicle has a low silhouette during waterborne operations, and the relative air velocity is low, the air drag can be neglected. If, however, the vehicle is a high-speed hydrofoil or a planing craft, the air drag becomes an appreciable portion of the total drag force and must be considered.

The drag due to the resistance of the water can be considered either as a total drag, or as the sum of its constituents; namely, the friction drag and the residual (viscous) or pressure drag. The total drag is that drag force which is required to overcome the total resistance of the water. This is the drag force which is normally obtained through model tests in the towing tank or water tunnel. The friction drag is that resistance to motion which is caused by the adhesion of the water to the surface of the vehicle. This drag can amount to nearly 95 percent of the total drag force in slow speed, nonstreamlined craft. The friction drag force can be obtained experimentally at low Froude numbers (essentially low speeds), or it can be obtained by theoretical means[159,160]. The pressure drag (also called the residual drag) is caused by differences in water pressure acting along the surface of the vehicle. It includes differences in pressure caused

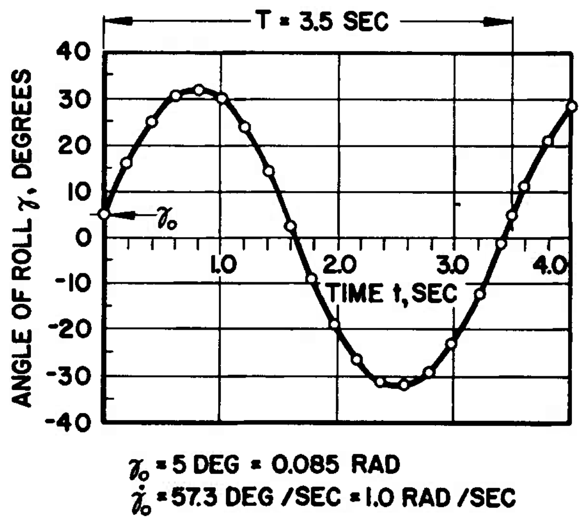


Figure 4-92. Rolling Motion As a Function of Time

by waves and eddies. The pressure drag is normally derived from model tests in a towing tank.

The present state of the art of hydrodynamic drag calculations are such that, if the proposed vehicle shape differs appreciably from a shape with known characteristics, any drag calculations based on theoretical estimates of drag coefficients are usually in error to the extent that they are useless. It is, therefore, suggested that the hull designer either base his waterborne shape on a very similar vehicle or rely on towing tank model testing as a means of obtaining drag estimates.

4-24.1 EFFECT OF SHAPE[163]

Total drag force (neglecting air drag) can be expressed as:

$$F_D = \frac{C_D \rho S V^2}{2} \qquad (4\text{-}128)$$

where

F_D = total drag force, lb
ρ = water mass density, slug/ft^3
S = wetted area of hull, ft^2
C_D = total drag coefficient, dimensionless
V = relative flow velocity, fps

This equation also applies to air drag, except S is the projected area of the vehicle exposed to the air.

Both the wetted area S and the drag coefficient C_D are functions of shape (C_D is influenced by other factors as well). The Taylor series of hull forms[164] approximates the wetted surface by the equation:

$$S = 2.6\sqrt{\bar{V}\ell} \qquad (4\text{-}129)$$

where

$\bar{V}$ = displaced volume of water, ft^3
ℓ = length of vehicle at waterline, ft

Figs. 4-93 and 4-94 illustrate the effect of shape on the drag coefficient C_D. In Fig. 4-93 it is noted that rounding the bow and stern has a significant effect on the magnitude of the drag coefficient at low velocities. At high velocities,

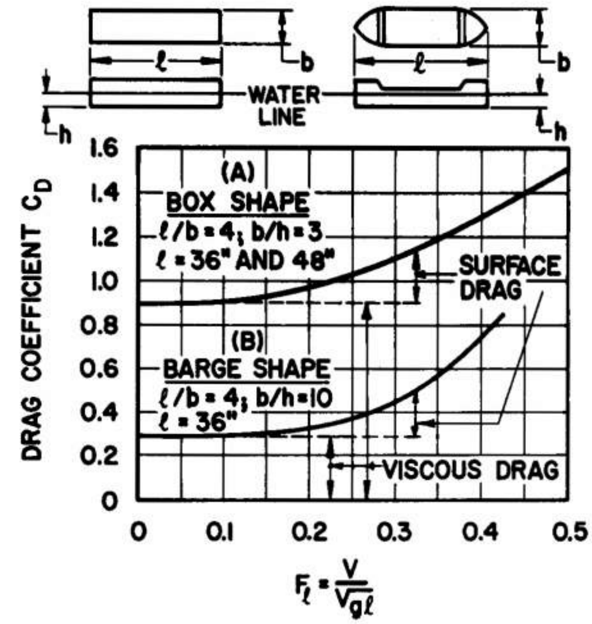


Figure 4-93. Drag Coefficients of Simple Barge Shapes[163]

the advantage of this particular shape is lost. This phenomenon is characteristic of displacement-type vessels which tend to produce similar drag coefficient levels at higher speeds. Fig. 4-94 shows the effect that rounding the forefoot of a simple barge-like shape has upon its drag coefficient. When making use of this figure, it is important to note that the data given are only applicable when the Froude number (see next paragraph) is between 0.14 and 0.25. Experimental data are lacking for other Froude members.

The symbol F_ℓ is the Froude number, named after William Froude (1810 to 1879) who was the founder of modern hydrodynamic model testing. The Froude number is an indication of the ratio of dynamic forces to the static forces and is expressed as:

$$F_\ell = \frac{V}{\sqrt{g\ell}} \qquad (4\text{-}130)$$

where

V = relative flow velocity, fps
g = gravitational constant, 32.17 ft/sec^2
ℓ = length of vehicle at waterline, ft

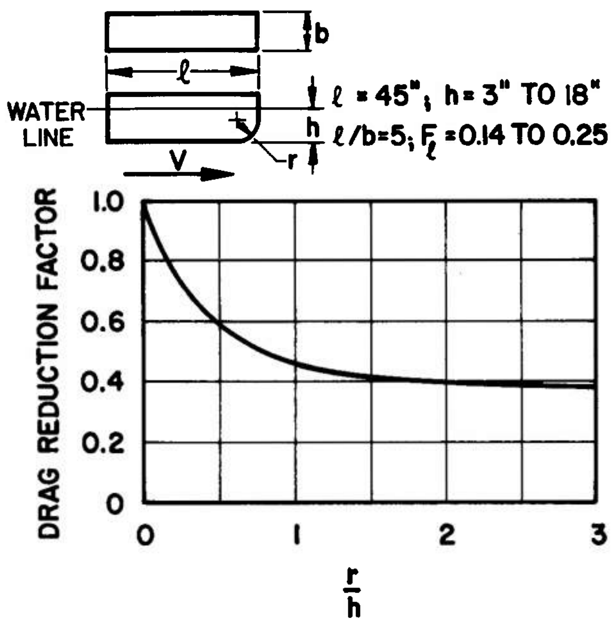


Figure 4-94. Effect of Forefoot Radius on Drag of Simple Barge Shapes[163]

4-24.2 EFFECT OF PROTRUSIONS[163]

Protrusions on the hull tend to increase the vehicle drag mainly by adding to its pressure drag. Principal protrusions include rudders, suspension gear, and the tracks or wheels. Fig. 4-95 illustrates the effect of shape on the drag coefficients of several submerged objects. The reduction in the drag coefficient by shape selection illustrates the need to streamline the protrusions as much as possible. The suspension system of amphibious tracked vehicles, for instance, can be covered by a side skirt to reduce the drag of the individual suspension system components.

The drag of protrusions are most easily analyzed by the use of towing tank or water tunnel models. This is particularly true for tracked vehicles in which the tracks are used for propulsion. The water flow near the surface of the vehicle is usually such that no protrusion is free from the influence of the turbulent water motion.

4-24.3 EFFECT OF VELOCITY[159,160,163]

From Eq. 4-128 it is apparent that the drag force is a function of the square of the velocity. It is not apparent, however, from the equation that the drag coefficient is velocity dependent. This effect is illustrated in Fig. 4-93 which indicates that the drag coefficient is relatively constant at low velocities (low Froude numbers) but then increases markedly after a certain velocity value. It can be generally stated, therefore, that the drag force is a function of the square of the velocity for low velocity values and is a function of both the velocity squared term and the nonconstant drag coefficient-velocity relationship for higher velocity values.

4-24-4 POWER REQUIREMENT[160]

The power requirement for propelling the vehicle is normally given in terms of effective horse-power *(EHP)*. This term is the power required by another vessel to two the given vessel at a particular speed. The effective horsepower is expressed by the equation

$$EHP = \frac{F_D V}{550} \tag{4-131}$$

where

550 = ft-lb/sec to horsepower conversion factor
F_D = total drag force, lb
V = relative flow velocity, fps

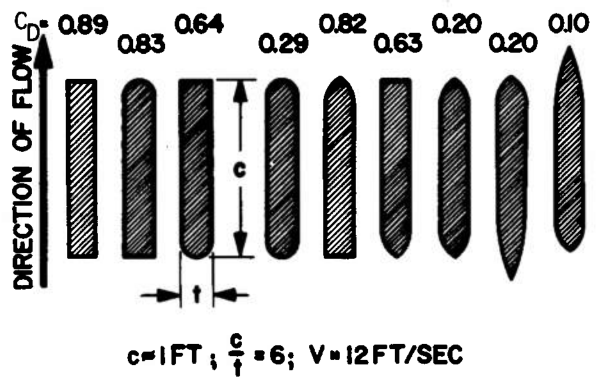


Figure 4-95. Drag Coefficients of Various Shapes in Two-dimensional Flow (Towed in Water Between End Plates)[163]

4-24.5 DESIGN CALCULATIONS

Consider again the illustrative vehicle shown in Fig. 4-89 and assume that it has no protrusions to cause additional drag.

Let the relative flow velocity V for this vehicle = 6 knots = 10.127 fps.

Total Drag Force

$$F_D = \frac{C_D \rho S V^2}{2}$$

$$\rho = \frac{64 \text{ lb/ft}^3}{32.17 \text{ ft/sec}^2}$$

$$\rho = 1.99 \text{ slug/ft}^3$$

$$S = 2.6\sqrt{\overline{V}\ell}$$

$$\overline{V} = \frac{\text{vehicle weight}}{\text{specific weight of water}} = \frac{35{,}000 \text{ lb}}{64 \text{ lb/ft}^3}$$

$$\overline{V} = 546.87 \text{ ft}^3$$

$$\ell = 20 \text{ ft}$$

$$S = 2.6\sqrt{(546.87)(20)}$$

$$S = 272 \text{ ft}^2$$

To obtain C_D we need the Froude number

$$F_\ell = \frac{V}{\sqrt{g\ell}}$$

$$F_\ell = \frac{10.127}{\sqrt{(32.17)(20)}}$$

$$F_\ell = 0.399$$

From Fig. 4-93(A), C_D at $F_\ell = 0.399$ is approximately equal to 1.27.

The total drag force is then:

$$F_D = \frac{(1.27)(1.99)(272)(10.127^2)}{2}$$

$$F_D = 35{,}251 \text{ lb}$$

Effective Horsepower Required

$$EHP = \frac{F_D V}{550} = \frac{(35{,}251)(10.127)}{550}$$

$$EHP = 649 \text{ hp}$$

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GLOSSARY

absolute filter. High efficiency particulate air filter characterized by having particle efficiencies better than 99.97 percent for 0.3-micron diameter particles as determined by MIL-STD-282, *Dioctyl Phthalate Tests.* Also referred to as "HEPA" filters.

absolute temperature. Temperature referenced to absolute zero which is approximately −459.6°F. If the temperature indicated by a common Fahrenheit thermometer is t and the absolute temperature is T, the absolute temperature is expressed as $T = t + 459.6$

acceleration. A vector quantity that expresses the time rate of change of velocity and is, thus, the first derivative of velocity and the second derivative of displacement of a moving body with respect to time.

Ackermann steering. The standard system of steering in which the front wheels are mounted on pivoted knuckles and are interconnected by a linkage. During a turn, the inner wheel pivots through a larger angle than does the outer wheel.

air delivery. A method of movement by air in which personnel, supplies, and equipment are unloaded from the aircraft while in flight. Cf. **air transportable.**

air landed. Personnel or cargo landed in aircraft as opposed to being dropped while the aircraft is in flight.

air transportable. **1.** State of being suited for transport by air. **2.** Capable of being air landed.

amphibious tank. *See:* **tank, amphibious.**

amphibious vehicle. A wheeled or track-laying vehicle capable of operating on both land and water.

angle of approach. The maximum angle of an incline onto which a vehicle can move from a horizontal plane without interference; as, for instance, from front bumpers.

angle of departure. The maximum angle of an incline from which a vehicle can move onto a horizontal plane without interference; as, for instance, from rear bumpers.

armor. Any physical protective covering, such as metals, plastics, ceramics, fabrics, and composites, used on military vehicles, aircraft, or persons against projectiles or fragments.

armored car. *See:* **car, armored.**

articulated steering. The system of steering used by tracked or wheeled vehicles consisting of two or more powered units in which the turning maneuver is accomplished by yawing the units with respect to each other about a pivot system not located over an axle of either unit.Cf. **wagon steering.**

articulated vehicle. A tracked or wheeled vehicle system consisting of two or more powered units.

athey wagon. A classification of heavy-duty, unpowered, tracked cargo trailers designed to be towed behind track-laying prime movers (tractors) for the transport of general cargo over soft or rough terrain.

automotive vehicle. A general category of mechanical land vehicles that contain a means of propulsion within themselves. They are generally considered to be either wheeled or track-laying; but, in the broad sense, this category includes all types of walking and jumping vehicles as well as self-propelled sleds and various air-cushion supported vehicles. They may have the ability to negotiate deep water barriers by swimming on the surface, in which case they are amphibious automotive vehicles, or by swimming submerged, in which case they are submarine automotive vehicles.

body. The assembly of panel sections, framing, doors, and windows of an automotive wheeled vehicle that form the main enclosure for cargo, passengers, and/or equipment. It excludes the entire chassis assembly and, in most cases, excludes the **cab.** Cf. **hull.**

bulkhead. Any type of partition wall that separates the space within a vehicle hull or body into compartments.

bulldozer. A broad, horizontal, metal, pushing blade with framework and equipment for mounting on and parallel to the front of a motorized vehicle; used for ground clearing and earth moving.

cab. The driver's compartment of a self-propelled vehicle.

car, armored. A wheeled self-propelled vehicle with protective armor plate designed for combat use and usually equipped with armament.

cargo tractor. *See:* **tractor, cargo.**

carrier, personnel. A self-propelled vehicle, sometimes armored, used for the transportation of troops and their equipment.

CBR. An abbreviation signifying "chemical, biological, and radiological".

center of pressure (CP). The point on a body at which all pressure forces acting on the body may be assumed to be concentrated; thus, it is the point at which the application of a single force will balance the summation of the pressure forces.

center of buoyancy (CB). The center of mass of the fluid displaced by a floating or submerged body. It is the point in a floating or submerged body at which the buoyant forces seem to be concentrated.

chassis. The undercarriage portion of a self-propelled wheeled vehicle consisting of a frame, power train, suspension system, controls—usually including the cab and hood—designed primarily to mount a suitable body for transporting cargo, personnel, and/or equipment.

combat tank, full-tracked. *See:* **tank, combat, full-tracked.**

combat vehicle. A land or amphibious vehicle, with or without armor or armament, designed for specific functions in combat or battle. The installation of armor or armament on vehicles other than combat vehicles does not change their original classification.

cross-country. Proceeding on a course directly over countryside, as across fields, hills, marshes, woods, and not by roads or paths.

deep fording. *See:* **fording.**

dolly. An auxiliary wheeled vehicle equipped with a drawbar, a fifth wheel assembly, brakes, springs, shock absorbers, wheels, tires, and other necessary equipment, designed to support the front end of a semitrailer and convert it into a full trailer.

durability. That characteristic, pertaining to an object, device, or system of devices, related to the period of satisfactory operation on a comparative basis. If two or more comparable items are subjected to the same operating conditions, the one that operates satisfactorily for the longest period is the more durable. Ability to withstand abuse is also a characteristic of a durable unit.

ethyl violet. A basic dye identified jointly by the American Association of Textile Chemists and Colorists and by the Society of Dyers and Colourists of Great Britain as Basic Violet 4, Chemcial Structure Type 11, Colour Index No. 42,600.

exoskeletal construction. A construction technique in which the body is a major stressed member. This is the principle of "unit construction" used by some automotive manufacturers and can result in a sizable reduction in vehicle weight.

external motion resistance. Resistance to motion of a vehicle in soil due to external forces such as bulldozing of the soil, soil compaction, and drag.

extraction provision. A provision integral to an item on its forward or aft end for use in attaching an extraction system during airdrop operations.

extraction system. A system of extraction parachute(s), extraction lines, necessary couplings, and load transfer devices used to withdraw air delivery items from aircraft in flight and, normally, to deploy the recovery parachutes.

fender. A guard or protective cover placed around the wheels or tracks of automotive vehicles to stop or deflect mud, water, stones, sand, and debris that are carried around by the wheels or tracks and that would otherwise be thrown about to create both a hazard and a nuisance.

fifth wheel. A flat round steel plate that is swivel-mounted on the frame siderails at the rear of a tractor truck and used for coupling a semitrailer. Part of a fifth wheel assembly.

fifth wheel assembly. A device for attaching a semitrailer to a truck tractor, or to a dolly, in such a way as to allow free rotation in a horizontal plane and yet prevent tipping.

fighting compartment. Portion of a fighting vehicle in which the occupants service and fire the principal armament. It occupies a portion of the hull and all of the turret, if any.

flat track suspension system. A suspension system on a tracked vehicle wherein the track returns on the top surfaces of the road wheels without the use of supplementary support rollers.

floating. This is the ability of a vehicle to negotiate water obstacles without being in contact with the bottom. Self-propulsion while in the water is not implied in this definition.

fording. This is the ability of a vehicle with its suspension in contact with the ground to negotiate a water obstacle of a specific depth. *Shallow fording* is fording without the use of special waterproofing kits, while *deep fording* is fording of greater depths with the application of a special waterproofing kit.

frame. The frame of an automotive assembly is a structure, separate from the body or hull, that supports the various components of the automotive assembly and maintains their spatial relationship. The frame provides strength and rigidity to the vehicle.

Goer-type vehicle. A four-wheeled vehicle having the following combination of distinguishing features: large diameter tires, exoskeletal construction, powered wagon wheel steering, power to all wheels, and suspension system consisting of tires only.

HEPA filter. *See:* **absolute filter.**

high velocity. As used in connection with artillery, small arms, and tank cannon, the term is generally accepted to having the following meanings: (1) a muzzle velocity of an artillery projectile of from 3,000 to, but not including, 3,500 fps; (2) muzzle velocities of small arms projectiles of 3,500 to 5,000 fps; and (3) velocities of tank cannon projectiles of 1,500 to 3,350 fps. Cf. **hypervelocity.**

hull. The body or hull of an automotive vehicle is the main structure that forms the passenger, cargo, and component compartments. The term "body" is usually applied to wheeled vehicles, while the term "hull" is applied to amphibious and tracked vehicles. Cf. **body.**

hypervelocity. As used in connection with artillery, small arms, and tank cannon, the term is generally accepted to have the following meanings: (1) muzzle velocities of artillery projectiles of 3,500 fps or higher; (2) muzzle velocities of small arms projectiles of 5,000 fps or higher; and (3) muzzle velocities of tank cannon projectiles in excess of 3,350 fps. Cf. **high velocity.**

idler. On track-laying vehicles, the wheel at the end of the vehicle opposite the driving sprocket, over which the track returns. It maintains track tension and reduces track skipping.

impedance. Mechanical impedance is the ratio of a force-like quantity to a velocity-like quantity when the arguments of the real (or imaginary) parts of the quantities increase linearily with time. Impedance is the reciprocal of mobility.

impulse. The product of a force and the time which the force is applied. More specifically, it is $\int_{t_1}^{t_2} F dt$ where F is time dependent and equal to zero before t_1 and after t_2.

impulse shock. A particular type of shock for which the waveform can be approximated by assuming a waveform of simple shape.

independent suspension. A system of arms, springs, wheels, etc., for elastically supporting the sprung mass of a vehicle, which permits the deflection of any one of the supporting wheels without substantially changing the load or position of the remaining wheels (as distinguished from solid axle or bogie suspension systems).

inertia resistance. As applied to an automotive vehicle, the inherent resisting forces opposing the linear and angular accelerations of the various masses of the vehicle.

internal friction. The portion of shear strength of soil that is proportional to the normal stress on the shearing surface. It is indicated by the term $p \tan\phi$ in Coulomb's equation: $s = c + p \tan\phi$.

internal motion resistance. Resistance to motion of a vehicle due to forces acting within and upon the vehicle such as friction between moving parts, hysteresis, inertia, vibrations, etc.

isolation. A reduction in the capacity of a system to respond to an excitation, attained by the use of a resilient support.

keying. The wedging of a projectile or fragment between two surfaces that normally are capable of relative motion so that no relative motion is now possible.

lunette. A towing ring or eye in the tongue or trail plate of a towed vehicle, such as a trailer or gun carriage, that is used in attaching the towed vehicle to the towing vehicle.

maintainability. The combined qualitative and quantitative characteristics of material design and installation which enable the accomplishment of operational objectives with minimum maintenance expenditures including manpower, personnel skills, test equipment, technical data, and facilities under the operational environmental conditions in which scheduled and unscheduled maintenance will be performed.

maintenance. All action taken to retain material in a serviceable condition or to restore material to serviceability. It includes inspecting, testing, servicing, classification as to serviceability, repairing, overhauling, modifying, modernizing, and rebuilding.

maximum working load. The maximum force which an item is expected to experience under normal use conditions.

metacenter. The point of intersection of the vertical through the center of buoyancy of a floating body with the vertical through a new center of buoyancy when the body is displaced in roll or pitch or both.

metacentric height. The distance between the metacenter of a body and its center of gravity.

Military Characteristics (MC). Those characteristics of equipment found desirable or necessary to the performance of a military mission, either combat or noncombat. Military Characteristics are prescribed by the using arms and usually form the basis of initiating development of a new item.

military vehicle. A wheeled or tracked vehicle specifically designed for military purposes including combat and/or the transporting of cargo, personnel, or equipment—or for the towing of other vehicles or equipment—overland and over roads in close support of fighting vehicles and troops. They are designed to withstand the rigors of the military environment, possess excellent cross-country performance capabilities, have built-in waterproofing to enable them to operate when submerged or have amphibious capabilities, and are provided with other special characteristics not normally provided in commercial-type vehicles.

Mimosa Z. A monoazo dye jointly identified by the American Association of Textile Chemists and Colorists and by the Society of Dyers and Colourists of Great Britain as Direct Yellow 9, Chemical Structure Type 3, Colour Index No. 19,540, and also known commerically as Naphthamine Pure Yellow G.

modulus of elasticity. The numerical value of the constant ratio of the unit stress within a material loaded in one direction to its corresponding unit strain within the proportional limit of the material.

modulus of resilience. The specific energy storing capacity of materials and expressed as in.-lb/in.^{3}

monocoque body. The ideal, theoretical limit of the integral frame and body vehicle construction. A single-shell body in which all parts are equally utilized as load bearing members, without the presence of a secondary structural layer of stiffeners or frame-like components. This structure is not realized but merely approached in automotive bodies. It is structure—the egg.

personnel carrier. *See:* **carrier, personnel.**

pintle assembly. A towing hook equipped with a hinged latch across its opening to retain the eye of a tow bar or lunette from becoming accidentally disengaged.

recovery system. A system of recovery parachute(s), riser extensions, load suspension slings, and parachute canopy ground release, and necessary coupling hardware used to retard and stabilize the descent of an airdropped item.

reliability. The probability of a device performing its purpose adequately for the period intended under the operating conditions encountered. For a system with independent components, the overall reliability is based on the product of the individual reliabilities; e.g., three independent components with a 90% reliability each, will have an overall reliability of 0.9X0.9X0.9 or 72.9%. Similary, 100 components with a 99% reliability each, will have an overall reliability of only 36.5%. Mechanical reliability as applied to military automotive equipment also includes the capacity of a vehicle to perform its mission after sustaining failure or destruction of specific components.

semitrailer. A nonpowered vehicle having integral wheels at the rear, only, and designed to carry materials, supplies, or equipment and to be towed by a self-propelled motor vehicle, that also supports the front end, by means of a fifth wheel coupling assembly. The front end can also be supported by a dolly that is provided with a fifth wheel assembly for coupling to the semitrailer and a tongue and lunette for coupling to the prime mover.

shallow fording. *See:* **fording.**

shear diagram. A graphic curve descriptive of a loaded beam wherein the abscissa represents distances along the length of the beam and the ordinate values represent the magnitudes and directions of the vertical shears at the various sections along the beam.

shroud. Rigid vertical plates of metal, rubber, and other material placed along the outboard sides of amphibious vehicles to enclose the upper portions of their tracks or wheels. Their purpose is to retain between the shrouds and the vehicle body the water carried around by the rotating tracks and thus effect a large hydraulic pump whose discharge is utilized to assist in the

propulsion of the vehicle.

stability. The property of mechanisms that cause them, when disturbed from conditions of static or dynamic equilibrium, to develop forces or moments to restore their original equilibrium conditions or, if the disturbance remains constant, to establish a new state of equilibrium.

steering system. The assembly of linkages and components which enables the driver to control the direction of the vehicle. Wheeled vehicles are normally steered by rotating the axes of rotation of two or more wheels with respect to the longitudinal center line of the vehicle, while tracked vehicles are usually steered by varying the speed of the tracks with respect to each other.

support roller. *See:* **track support roller.**

suspension provision. An integral part of an item of equipment used as a means of attaching a load suspension sling.

suspension system. The mechanical linkages and the elastic members that provide a flexible support for the sprung components of a vehicle.

swimming. The ability of a vehicle to negotiate a water obstacle by propelling itself across, without being in contact with the bottom.

tactical vehicle. Any vehicle designed for field requirements in combat and tactical operations, or for training personnel for such operations.

tank, amphibious. Vehicle mounting a howitzer or cannon, capable of delivering direct fire from the water as well as ashore, and used in providing early artillery support in amphibious operations.

tank, combat, full-tracked. A self-propelled, heavily armored, offensive vehicle having a fully inclosed revolving turret with one major weapon. It may mount one or more machine guns. Excludes self-propelled weapons.

tank transporter. Special-purpose wheeled or tracked vehicle, or combination of vehicles, designed to transport tanks or other heavy vehicles over highway and natural terrain, and incorporating integral provisions for loading and unloading disabled vehicles without supplemental assistance.

tiedown provision. An integral part of an item of equipment such as a pad with an eye, shackle, ring, or other device for use in attaching a tiedown lashing or other device to the item.

top roller. *See:* **track support roller.**

torr. A unit of pressure equal to 1 mm Hg at 0°C. One standard atmosphere is equal to 760 torr.

track. The continuous band or segmented chain upon which a tracked vehicle runs. Cf. **track-laying vehicle.**

track-laying vehicle. A vehicle that utilizes endless belts or tracks to distribute its gross load over the supporting ground to achieve more uniform ground pressure for improved traction and mobility on adverse soils.

track support roller (top roller) (return roller). One of a number of wheels that support the top run (return run) of the track between the driver sprocket and idler of a track-laying vehicle.

tractor. A track-laying vehicle designed to tow by means of a pintle hook or fifth wheel coupling device.

tractor, cargo. Military track-laying vehicle designed to carry cargo, as well as to perform as a tractor.

tractor truck. A short wheelbase wheeled vehicle designed to tow and partially support a semitrailer through a fifth wheel coupling device.

trailer. A wheeled or tracked vehicle, nonpowered, with all or most of its weight supported by its own integral wheels or tracks, designed to carry materials, supplies, or equipment and to be towed by a self-propelled motor vehicle. Excludes **semitrailer.**

transport vehicle. Vehicle primarily intended for personnel and cargo carrying. Excludes **combat vehicle.**

transportability. The capability of an item of military equipment to be transported efficiently and effectively via railways, highways, waterways, oceans, and airways, either by carrier, by being towed, or by self-propulsion.

truck. A self-propelled wheeled vehicle designed primarily to transport supplies and/or equipment and which may be used to tow trailers or other mobile equipment. Excludes **truck tractor.**

truck tractor. A self-propelled wheeled vehicle designed to tow and partially support a semitrailer by means of a fifth wheel-type coupler.

turret. An enclosed, cylindrical or dome-shaped structure, usually armored and rotatable, into which are mounted one or more guns or similar weapons in a manner that protects their breech portions and loading mechanisms and which

may also house and protect the weapon crew.

ultimate strength. The maximum load required to produce failure of a stressed item due to physical rupture or breakage.

unitized body. A type of vehicle construction in which the frame and body floor are integrated into a single structure characterized by a number of formed, ribbed, and reinforced sheet metal panels and few well placed structural shapes welded into a unit.

vertical shear. The magnitude of the resultant of the vertical forces (loads and reactions) that lie on one side of a particular section of a beam. It is the algebraic sum of the vertical forces that lie on one side of a beam section being investigated.

wagon steering. Steering of a vehicle consisting of one or more units by a single pivot system with the pivot point located over the front axle. Cf. **articulated steering.**

yield strength. The load which produces a permanent deformation or set in a stressed item, in the direction of the applied load.

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